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L3	L1 and ((current adj position adj5 actuator) and (calculat$3 adj3 correct$3 adj3 (value or parameter)))	0	L3
L2	L1 ((current adj position adj5 actuator) and (calculat$3 adj3 correct$3 adj3 (value or parameter)))	418	L2
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END OF SEARCH HISTORY

L1: Entry 143 of 413 File: USPT Apr 25, 2000

DOCUMENT-IDENTIFIER: US 6055473 A
TITLE: Adaptive seating system

Abstract Text (1):
Adaptive seating comfort control automatically senses occupation of a seat, estimates attributes of the seat occupant, and generates and issues seating comfort control commands in response to the occupant attributes to provide for maximum seating comfort and to provide for a desired interaction with other control systems. Comfort control commands take the form of desired pressure commands for at least one fluid filled bladder within the seat and may be updated periodically during a period of seat occupation to account for change in position of the seat occupant or to reduce occupant muscle fatigue.

Brief Summary Text (2):
This invention relates to automotive seating and, more particularly, to automatic seat comfort control.

Brief Summary Text (4):
Manually-controlled automotive seating systems are known. Such systems may include seating firmness control for seat occupant comfort in which the occupant manually controls actuators for varying fluid pressure in one or more bladders within a seat. To assure occupant comfort, the fluid within each bladder must be set to a pressure corresponding to the physical attributes of the seat occupant and to the occupant's position within the seat. In a seating system having a plethora of such bladders, the manual process of bladder pressure adjustment can be difficult and time consuming. Unless the occupant takes the time to properly adjust all bladders, the seat may be uncomfortable. As a single occupant shifts in position within the seat, the pressure requirements within the bladders may change, requiring further manual input to assure seating comfort. Over an extended driving period, occupant comfort requirements may change and an initial fluid pressure setting within one or more of the bladders may require updating. Occupant muscle fatigue and discomfort may result unless proper manual pressure adjustment is provided. Manually controlled automotive seating systems may further include manually controlled actuators for positioning or orienting the seat. Unless the seat position or orientation corresponding to the attributes of the seat occupant and the position of the seat occupant within the seat is identified and maintained throughout a period of occupation of the seat, occupant comfort may be reduced and proper interaction of the seating system with other systems may not be assured.

Brief Summary Text (5):
It would therefore be desirable to provide an automotive seating control system that assures proper seating control with minimum burden placed on the seat occupant to establish and maintain proper settings of a seating control system.

Brief Summary Text (7):
The present invention is directed to a seating control system that automatically identifies seat occupant requirements and adjusts the seat accordingly to maximize occupant comfort and to assure proper

interaction with other systems.

Brief Summary Text (8):
More specifically, a vehicle seating control system includes a plurality of bladders secured within a seat and a fluid pressure control system for controlling passage of a fluid, such as air, into and out of the plurality of bladders. At least one pressure transducer is secured within the seat and transduces pressure applied to the seat into a signal passed to control circuitry. The control circuitry periodically samples the pressure signals applied thereto and estimates attributes, such as corresponding to occupant height or weight and occupant orientation within the seat in response to the signal samples. Seating system adjustments are made in response to the estimated occupant information to yield a desirable seat orientation to maximize comfort and to support a desired interaction with other vehicle systems.

Brief Summary Text (9):
In accord with a further aspect of this invention, after providing for an initial control setting of the seating system, the control circuitry periodically samples the signal values to determine if any change in occupant position or orientation within the seat has occurred. If a change has occurred, adjustments are determined and are made to maintain a desirable seat orientation throughout a period of occupation of the seat. In accord with yet a further aspect of this invention, the time of occupation of the seat is monitored. If such time exceeds a threshold time corresponding to an extended period of occupation of the seat, a slight adjustment of the seat orientation is automatically made to reduce occupant muscle fatigue. The threshold time is identified as the typical time of occupation after which muscle fatigue is likely. Such adjustments may be periodically provided throughout a period of occupation of the seat and, over an extended period of occupation, may become more frequent as a propensity for occupant muscle fatigue increases. The seat orientation information may be stored for re-use from one period of seat occupation to the next.

Brief Summary Text (10):
In accord with yet a further aspect of this invention, the seating adjustments automatically made in response to estimated occupant attributes or orientation within the seat include seat positioning adjustments. In a seating system in which a headrest is positioned adjacent a seatback, such adjustments may take the form of variation of the position of the headrest relative to the position of the seatback.

Brief Summary Text (11):
In accord with a further aspect of this invention, the sensors take the form of pressure transducers disposed within pneumatic control lines opening into sealed bladders positioned at various locations within the seat. The bladders are maintained at a nominal pneumatic pressure. Occupation of the seat is detected when the sensor output signals indicate a significant increase in pressure across the bladders, which initiates a determination of occupant information and automatic control of the pressure within the bladders, such as through a series of valved pneumatic lines for passing pressurized air to and from the bladders. In accord with still a further aspect of this invention, manual override of the automatic seating control is provided immediately upon a manual control input through a seat occupant interface. Automatic control is resumed when manually requested by the seat occupant.

Drawing Description Text (3):
FIG. 1 is a perspective view of an automotive vehicle seat with an installation of the automatic seat control system of this embodiment, the seat control system being schematically shown;

Drawing Description Text (4):
FIG. 2 is a general block diagram of the seat control hardware of the seat control system of FIG. 1; and

Drawing Description Text (5):
FIGS. 3 and 4 are flow diagrams illustrating a control sequence for carrying out automatic seat control operations through the hardware of FIG. 2.

Detailed Description Text (2):
Referring to FIG. 1, an automotive vehicle seat 10 includes a seatback 12 secured to a seat cushion 14 in any suitable conventional manner. A plurality of sealed bladders 20-38 are retained within the seatback 12 and seat cushion 14 in any suitable conventional manner and a pneumatic pressure control system operates, in accord with principles of this invention, to automatically vary the air pressure within the bladders. The series of bladders of this embodiment include left and right seat cushion wing bladders 20 and 22, respectively, within corresponding left and right seat cushion wings 14a and 14b, respectively, within respective left and right seatback wings 12a and 12b, left and right thigh support bladders 26 and 24, respectively, butt pocket area bladder 28, left and right seatback wing bladders 32 and 30, respectively, within respective left and right seatback wings 12a and 12b, and upper, middle and lower lumbar support bladders 34, 36, and 38, respectively. A headrest 16 is positioned adjacent the seatback 12 at an upper end thereof and is mechanically linked to an output shaft (not shown) of a linear actuator 54. The linear actuator 54 takes the form of a conventional DC motor or stepper motor secured within the seatback 12 and is electrically driven to raise and lower the headrest 16 relative to the seatback 12 in response to control command Cpos issued by a controller 60 through a standard motor driver interface 62. A linear position transducer 56, such as in the form of a standard linear potentiometer, is installed adjacent the actuator 54 to transduce the position of the actuator output shaft into output signal P.sub.HR. For example, the transducer 56 may include a wiper arm that slides along a resistive, electrically conductive track of the transducer as the actuator output shaft is raised and lowered, with a constant electrical potential applied across the resistive track, whereby the electrical resistance between one end of the conductive track and the wiper arm is proportional to the position of the headrest 16, as is generally known in the art.

Detailed Description Text (3):
The controller takes the form of a single chip microcontroller, such as a commercially available Motorola MC68HC11 microcontroller. The controller 60 receives the transducer output signal P.sub.HR from the position transducer 56 and receives pressure signals Prs from standard pressure transducers disposed within pneumatic control lines that open into the bladders 20, 24, 26, 28, 30, 34, 36, and 38, or directly within the bladders 20-38 themselves, only one pressure transducer 40 being shown in FIG. 1. Bladders 20 and 22 and bladders 30 and 32 share a common pressure transducer as they open into and therefore are maintained at substantially a common pressure, as will be further detailed in FIG. 2. A plurality of conventional seat position transducers, for example of the potentiometric type, are secured to the seat position control assembly, which may take any conventional form including either manual or power control capability. The seat position transducers provide output signals indicating the orientation of the seat 10, including the fore/aft seat position, the angle of the seatback 12, and the lift position of both the front 15 and rear 17 of the seat cushion 14. The manner of transducing and indicating such seat position information is generally understood in the seating control art. The seat position signals, indicated generally as seat position signal Pseat (FIG. 1), are provided as inputs to controller 60. A keypad 66 of any suitable standard form transduces command information manually provided by an operator (such as a seat occupant) into control signals passed to the controller 60 and to bladder pressure control hardware, generally illustrated in FIG. 1 by block 64. The keypad 66 may include any combination of suitable standard buttons, levers, dials, or any other conventional apparatus known in the art to transduce manual commands into electrical signals and is positioned in the vehicle interior in position to be easily accessed by the vehicle operator. The bladder pressure control hardware 64 includes a combination of pumps, valves, and control circuitry for individually controlling pressure within the bladders 20-38 in response to either manual commands from the operator via the keypad 66 or automatic

control commands PC issued by the controller 60, with manual commands having control priority to override any automatic commands in this embodiment, as will be described further.

Detailed Description Text (4):
Referring to FIG. 2, further details of the pressure control system of this embodiment are illustrated. Generally, manual control inputs from an operator (such as a seat occupant) via keypad 66 are given a higher control priority than the automatic control commands generated by controller 60, whereby automatic control operations will immediately cease upon receipt of any manual input via the keypad 66 and will not resume unless an appropriate key on the keypad 66 is depressed, such as a key labeled "AUTOMATIC CONTROL" or a like legend. The keypad 66 includes at least two discrete output signals AC and INT. AC is set to an active signal level when automatic control is requested by the operator and INT changes state when manual control is initiated by an operator, such as by applying a manual input to the keypad 66. The signal AC is a digital signal applied to a standard input port of the controller 60 which is periodically polled through controller operations to determine whether automatic control is desired, as will be further detailed. The signal INT is an interrupt signal received at a standard input port of the controller 60 to trigger an interrupt of controller operations when the signal changes state.

Detailed Description Text (5):
The bladder pressure control hardware 64 of FIG. 1 is further detailed in FIG. 2 and generally includes a driver 76, seat cushion and seat back drive modules, 72, and 74, respectively, pumps 78 and 80, and pneumatic control lines 82, 84, 178, 180, and 220-238. The keypad 66 is directly coupled to the driver 76 which includes standard control and drive circuitry for selectively activating and driving first and second air pumps 78 and 80, and for controlling the seat cushion valve module 72 and the seatback valve module 74. The pumps 78 and 80 are commercially available air pumps of any suitable form. The seat cushion valve module 72 includes a standard valve body for receiving pressurized air from air pump 78 via supply line 178 and opening to the atmosphere via exhaust line 82 and, through a series of standard electronically controlled valves of the solenoid, butterfly, or other conventional type, for selectively coupling the supply line 178 or exhaust line 82 to any of the pressure control lines 220-228.

Detailed Description Text (6):
The pressure control lines 220-228 open into bladders in the seat cushion. More specifically, control line 220 opens into bladders 20 and 22, control line 224 opens into bladder 24, control line 226 opens into bladder 26, and control line 228 opens into bladder 28. Pressurized air in the supply line 178 is passed through the seat cushion valve module 74 to any bladder requiring an increase in air pressure therein up to a pressure limit, as commanded manually via keypad 66 or as commanded through the operations of controller 60. In the event a reduction in air pressure in any of the bladders 20-28 is commanded manually via the keypad 66 or automatically via the controller 60, the seat cushion valve module 72 opens a passage between the control line of such bladder to the exhaust line 82 to allow the pressurized air to be exhausted to the atmosphere. Pressure transducers 120, 124, 126, and 128 are disposed within respective control lines 220, 224, 226, and 228, to transduce the air pressure in the respective control line and therefore in the corresponding bladder into an output signal which is applied to input ports of the controller 60. The driver 76 generates control commands for controlling the position of the valves within the seat cushion valve module 72 and communicates the commands via a standard communication link 172.

Detailed Description Text (8):
Pressurized air in the supply line 180 is passed through the seat back valve module 74 to any bladder requiring an increase in air pressure therein up to a pressure limit, as commanded manually via keypad 66 or as commanded through the operations of controller 60. In the event a reduction in air pressure in

any of the bladders 30-38 is commanded manually via the keypad 66 or automatically via the controller 60, the seatback valve module 74 opens a passage between the control line of such bladder to the exhaust line 84 to allow the pressurized air to be exhausted to the atmosphere. Pressure transducers 130, 134, 136, and 138 are disposed within respective control lines 230, 234, 236, and 238 to transduce the air pressure in the respective control lines and therefore in the corresponding bladder into an output signal which is applied to input ports of the controller 60. The driver 76 generates the control commands for controlling the position of the valves within the seatback valve module 74 and communicates the commands to the module 74 via a standard communication link 174.

Detailed Description Text (9):
The controller 60 issues a position control command for controlling the spacing between the headrest 16 (FIG. 1) and the seatback 12 and applies the control command to the actuator 54 via a standard motor driver 62. The displacement of the actuator output shaft (not shown), which is secured to the headrest 16 of FIG. 1 is transduced by conventional displacement sensor 56 into output signal P.sub.HR which is applied to a standard controller input port for use in closed-loop control of the headrest position. Likewise, the controller 60 outputs seat bladder pressure control commands to the driver 76 for varying the pressure within the bladders 20-38 when automatic pressure control operations are active. The seat position signal information is provided directly from the described conventional position transducers (not shown) to the controller 60.

Detailed Description Text (12):
Returning to step 304, if the flag is determined to be set, automatic control operations begin by sampling the pressure input signals from the pressure transducers 120-138 (FIG. 2), the seat position signals Pseat and the position signal P.sub.HR indicating headrest 16 (FIG. 1) height at a step 306. If the pressure signals indicate a substantial pressure increase above predetermined nominal pressure values, wherein the nominal pressure is determined through a standard calibration process, then the seat is assumed to be occupied at a next step 308. Otherwise, the seat is assumed to be vacant at the step 308, and a step 309 is then executed to clear memory values corresponding to the prior occupant, including a seat occupied flag and stored values indicating the height and weight of the prior occupant, and desired pressure values and a desired headrest height value. After the step 309, the sampling operations of the step 306 are repeated, and the sampled values analyzed at the step 308 to determine if the seat is occupied. The steps 306, 308, and 309 are repeated until occupation is detected at the step 308. Upon detecting occupation of the seat, a prior occupation flag is next polled at a step 310. If the prior occupation flag is set, then the seat has been occupied for at least the last two iterations of the routine of FIG. 3, and the control operations of steps 338-348 are carried out to determine if any pressure adjustments are currently necessary. If the prior occupation flag is not determined to be set at the step 310, then the current iteration of the operations of FIG. 3 is the first since the current set occupant has been detected, and seat setup operations of steps 312-336 are required, beginning by setting the prior occupation flag at the step 312, and storing the current value of any available free running clock in the controller 60 (FIG. 2) to a memory location at a step 314. The occupant's height and weight are next estimated as a function of the pressure signal and seat position samples taken at the most recent prior execution of the operations of step 306. In this embodiment, within the general layout of the bladders 20-38 and corresponding pressure transducers 120-138 described for FIGS. 1 and 2, the occupant's height H is estimated in millimeters as follows:

Detailed Description Text (13):
in which .DELTA.P.sub.CW is the pressure difference between the current pressure and a predetermined nominal pressure, termed the "vacant pressure" of the seat cushion wing bladders 20 and 22 (FIG. 1), .DELTA.P.sub.LT is the pressure difference between the current pressure and the "vacant pressure" of the left thigh support bladder 26 (FIG. 1), .DELTA.P.sub.UL is the pressure difference between the

current pressure and the "vacant pressure" of the upper lumber bladder 34 (FIG. 1), .DELTA.P.sub.LL is the pressure difference between the current pressure and the "vacant pressure" of the lower lumber bladder 38 (FIG. 1), .DELTA.P.sub.BP is the pressure difference between the current pressure and the "vacant pressure" of the butt pocket bladder 28 (FIG. 1), .DELTA.P.sub.RT is the pressure difference between the current pressure and the "vacant pressure" of the right thigh support bladder 24 (FIG. 1), and P.sub.FA is the transduced "fore/aft" position of the seat 10 (FIG. 1). The "vacant pressure" is the pressure transduced by a pressure transducer under predetermined nominal conditions, such as while the seat is vacant. The constant K1 and the coefficients C1-C7 may be determined through a conventional calibration process in which the relationship between the pressure distribution about the seat 10 (FIG. 1) and the height of the occupant is established.

Detailed Description Text (15):
The constant K2 and the coefficients C11-C17 may be determined through a conventional calibration process in which the relationship between the pressure distribution about the seat 10 (FIG. 1) and the weight of the occupant is established.

Detailed Description Text (16):
Returning to FIG. 3, the estimated height and weight are next stored in memory at a step 318. In an alternative embodiment within the scope of this invention, the controller 60 periodically outputs the determined height H and weight W information on a standard communication link (not shown) to other vehicle control or diagnostic systems that may benefit from such information, such as chassis or suspension control systems. Such other control systems may then take action to adjust control commands or diagnostic information in response to the occupant information to better provide for vehicle control or diagnostics, for example by adjusting the vehicle ride softness to most appropriately accommodate the current seat occupant. In an embodiment of this invention in which occupant height and weight information for is estimated through the operations of FIG. 3 for each seat occupant within the vehicle, and such information is provided to other control systems or to diagnostic systems through a standard communication link (not shown), comprehensive control or diagnostic enhancements to such other systems may be provided.

Detailed Description Text (18):
in which CP is the current bladder pressure reading. More specifically, the individual bladder pressure change values are expressed in expanded form broken down into the individual components of the occupant's height and weight calculation terms, as follows. ##EQU1## In which .DELTA.PC.sub.LT is the commanded change in pressure in the left thigh support bladder 26 (FIG. 1), expressed in volts, P.sub.FC is the transduced lift position away from a base lift position of the front 15 of the seat cushion 14, .DELTA.P.sub.ML is the pressure change away from a calibration pressure of the middle lumber support bladder 36 (FIG. 1), K3 is a calibration constant, and C21-C29 are calibration coefficients. ##EQU2## In which .DELTA.PC.sub.RT is the commanded change in pressure in the right thigh support bladder 24 (FIG. 1), expressed in volts, .DELTA.P.sub.BW is the change in pressure of the backwing support bladders 30 and 32 away from a "vacant pressure," P.sub.SB is angle of the seatback 12 (FIG. 1) relative to a predetermined angle, K4 is a calibration constant, and C31-C39 are calibration coefficients. ##EQU3## In which .DELTA.PC.sub.BP is the commanded change in pressure in the butt pocket support bladder 28 (FIG. 1), expressed in volts, EXP[] is notation for the exponential function, P.sub.RC is the lift position of the rear section 17 of the seat cushion 16 (FIG. 1), K5 is a calibration constant, and C40-C413 are calibration coefficients.

Detailed Description Text (19):
In which .DELTA.PC.sub.CW is the commanded change in pressure in the seat cushion wings support bladders 20 and 22 (FIG. 1), expressed in volts, K6 is a calibration constant, and C50-C57 are calibration

coefficients.

Detailed Description Text (21):
.DELTA.PC.sub.BW is the commanded change in pressure in the seat back wing bladders 30 and 32 (FIG. 1), expressed in volts, and is set to about seven volts in this embodiment in which seatback 12 is sufficiently wide that pressure adjustments in the seat back wings 30 and 32 are not detectable for most small and medium sized occupants, so a constant pressure value consistent with comfort for large occupants is maintained.

Detailed Description Text (23):
After determining the headrest height command at the step 322, the pressure change commands are output to the driver 76 (FIG. 2) at a next step 328 to initiate activation of the applicable pumps 78 and 80 (FIG. 2) and to direct the valve modules 72 and 74 as necessary to vary pressure in the applicable bladders 20-38 in accordance with the pressure change commands. The headrest height command is next output to the driver 62 (FIG. 2) at a step 334 to drive the actuator 54 in direction to provide the desired headrest height. The current value of a controller clock taking any standard form is next stored in a controller memory device at a step 336 to mark the time of the adjustment of the seat 10 (FIG. 1). The operations of FIG. 3 are next concluded by returning, via a next step 336, to resume execution of any previously ongoing and temporarily suspended controller operations, such as standard maintenance and diagnostic operations generally understood by those possessing ordinary skill in the art.

Detailed Description Text (24):
Returning to step 310, if the prior occupation flag is determined to be set, then a determination is made of the time T.sub.A elapsed since the most recent prior seat adjustment at a next step 338. If the determined time T.sub.A exceeds a predetermined threshold which may be calibrated within the range of 0.01 seconds to 3600 seconds in this embodiment, with a typical value of about sixty seconds, at a next step 340, then a seat adjustment to account for any change in occupant position or orientation within the seat 10 (FIG. 1) or to reduce operator muscle fatigue is provided via steps 342-348. More specifically, a series of bladder pressure adjustment commands are determined at a step 342. Such adjustments may be made as a function of the current sampled seat bladder pressure information from the described step 306, for example in the manner detailed at the described step 320, or may consist of slight pressure increases or decreases in the bladders 20-38 (FIG. 2) of any type that will relieve pressure on certain operator muscle groups and generally provide for increased comfort over extended driving periods.

Detailed Description Text (26):
Referring to FIG. 4, a series of controller operations to be carried out in this embodiment following any manual operator input of a seating attitude change command via the keypad 66 of FIG. 2 are illustrated. Generally, automatic seating control is provided when requested by an operator, such as a seat occupant, through depression of an automatic control key on the keypad 66 (FIG. 1). Such automatic control is terminated upon any seat bladder pressure change command via the keypad 66. A signal INT is provided from the keypad 66 (FIG. 2) to a standard controller input port and changes state upon any manual seating control input on the keypad. The applicable controller input port is configured to treat such change in state as an interrupt event, upon which event the controller 60 (FIG. 2) is directed to suspend its current operations and to carry out the operations of FIG. 4, beginning at a step 400 and proceeding to deactivate automatic control at a next step 402, such as by clearing the automatic control flag polled through the operations of FIG. 3 at the described step 304. The current set of bladder pressure values are stored as the current desired bladder pressure values at a next step 404 as a record of the current commanded pressure values for the bladders 20-38 (FIG. 2) whereby such values may be used to quickly return to the current seat bladder pressure values upon exiting the manual seat bladder pressure control mode. Further, the current headrest height is stored at a next step 406 as a value to return to upon

re-entry of the automatic control mode of operation. The operator has the authority to manually request re-entry into the automatic seat control mode in this embodiment. Following the step 406, the interrupt service operations of FIG. 4 are concluded by returning via a next step 408, to any standard controller operations that were temporarily suspended to allow for execution of the operations of FIG. 4.

CLAIMS:

1. An automatic seating comfort control system, comprising:

a seat having a plurality of bladders disposed therein;

a fluid pressure control system for controlling passage of a fluid into and out of selected ones of the plurality of bladders;

at least one pressure transducer for transducing fluid pressure within the fluid pressure control system into at least one corresponding pressure signal; and

a controller for receiving the pressure signal and for estimating seat occupant dimension information as a function of the received pressure signal, the occupant dimension information including at least one of occupant height and weight information;

the controller generating seating control signals as a function of the seat occupant dimension information and outputting the seating control signals to the fluid pressure control system to automatically control seating comfort.

2. The system of claim 1, wherein the seat is secured within an automotive vehicle having a plurality of vehicle control systems, the automatic seating comfort control system further comprising:

a communication link for communicating the seat occupant dimension information to at least one of the plurality of vehicle control systems.

5. The system of claim 3, further comprising:

at least one seat position transducer for transducing seat orientation into at least one corresponding position signal;

and wherein the controller further receives the position signal and estimates seat occupant dimension information as a function of the received pressure signal and the received position signal.

6. The system of claim 3, wherein the seating control signals generated and output by the controller include at least one desired seat position control signal for controlling seating position.

7. The system of claim 6, wherein the seat includes a seatback and a headrest adjacent the seatback, the system further comprising:

an actuator secured within a first one of the group consisting of the seatback and the headrest, the actuator having an output shaft coupled to a second one of the group consisting of the seatback and the headrest, the actuator controlled in response to a headrest position control signal applied thereto to extend and retract the output shaft relative to the actuator to vary the position of the headrest relative to the seatback;

and wherein the at least one desired seat position control signal includes the headrest position control signal.

8. The system of claim 1, further comprising:

an operator interface for transducing a request for manual seating control from an operator into a manual control request signal and for transducing manual control commands from an operator into manual seat control command signals;

wherein the fluid pressure control system receives the manual seat control command signals and controls passage of a fluid into and out of selected ones of the plurality of bladders in response to the received manual seat control command signals;

and wherein the controller receives the manual control request signal and suspends automatic seating control operations upon receipt thereof.

9. A method for controlling firmness of an automotive vehicle seat having a plurality of fluid-filled bladders disposed therein and having a fluid pressure control system for varying the fluid pressure in individual ones of the plurality of fluid-filled bladders in response to pressure control commands, comprising the steps of:

transducing pressure within the fluid pressure control system into corresponding pressure signals;

periodically sampling the pressure signals;

sensing presence of an occupant within the seat;

upon sensing the presence, estimating attributes of the occupant as a function of the sampled pressure signals, the attributes including at least one of height and weight of the occupant;

generating desired pressure values for at least one of the plurality of fluid-filled bladders as a function of both the sampled pressure signals and the estimated attributes of the occupant;

determining pressure control commands as a function of the desired pressure values; and

controlling pressure within at least one of the plurality of fluid-filled bladders in accordance with the determined pressure control commands.

10. The method of claim 9, the seat having a headrest adjacent thereto and an actuator for varying the position of the headrest relative to the seat in response to a headrest position control command, the method further for controlling automotive seat headrest position and further comprising the steps of:

calculating a desired headrest position as a function of the occupant attributes;

generating a headrest position command as a function of the desired headrest position; and

controlling headrest position in accordance with the generated headrest position command.

11. The method of claim 9, wherein the automotive vehicle includes an additional vehicle control

system, the method further comprising the step of:

communicating the estimated attributes of the seat occupant to the additional vehicle control system.

L1: Entry 84 of 413 File: USPT Jul 9, 2002

DOCUMENT-IDENTIFIER: US 6415814 B1
TITLE: Vibratory patient support system

Brief Summary Text (31):
Preferably, the low air loss patient support system is divided into sections. Control means are provided for maintaining the sacs within each section at a particular predetermined pressure by computing and maintaining a height and weight specific pressure profile for each section. The vibrational forces imparting means is independently actuable and controllable relative to the control means for maintaining the sacs at a predetermined pressure. In this embodiment, the low air loss patient support system can function regardless of whether the vibrational system is actuated. On the other hand, actuation of the vibrational force system in no way degrades or effects the low air loss aspect of the patient support system.

Brief Summary Text (38):
The means for supplying air to each sac further preferably includes a support member carried by the frame. The support member preferably is rigid to provide a rigid carrier on which to dispose the sacs and may comprise a plurality of separate non-integral sections so that a one-to-one correspondence exists between each support member section and each articulatable section of the frame. Each section of the rigid support member preferably comprises a modular support member that defines a multi-layered plate which has an upper layer, a lower layer and a middle layer between the other two. The three-layered plate has a top surface, a bottom surface, two opposed ends, and two opposed side edges. A plurality of inlet openings are defined through at least one of the side edges. In appropriate embodiments, a plurality of exit openings are defined in the opposite side edge. For example, the plate at each end of the patient support only has inlet openings defined through one of the side edges. A plurality of air sac supply openings are defined through the plate from the top surface and preferably extend completely through the three layers of the plate. In at least one of the plates, preferably the seat plate, a plurality of pressure control valve openings are defined through the bottom surface of the plate. A plurality of channels preferably are defined and enclosed between the top surface and the bottom surface of the plate and connect the various inlet openings, outlet openings, air sac supply openings, and pressure control valve openings to achieve the desired configuration of air supply to each of the sacs disposed atop the top surface of the plate.

Detailed Description Text (13):
As shown schematically in FIG. 13 for example, at least one of the modular support members defines a seat sack support member 94 (Zone III) and includes a plurality of pressure control valve openings 96 defined through the lower layer 76 and extending through the bottom surface of the plate 70. Each pressure control valve opening 96 is configured to be connected to a pressure control valve (described hereinafter). Each of the ten pressure control valve openings 96 shown in FIG. 13 is schematically represented by a circle inscribed within a box. To avoid unnecessarily cluttering FIG. 13, only three of the pressure control openings are provided with designating numerals 96. Preferably, one end of a rigid

elbow 98 (FIGS. 7 and 8) has a flexible bellows (not shown) which is connected to each pressure control valve opening 96, and the other end of the elbow is connected to the output end of the pressure control valve. The seat sack support member preferably includes at least one pressure control valve opening for each pressure control valve required by the particular configuration of the patient support system. Each pressure control valve opening intersects with a channel (described hereafter) for supplying air to the air sacks.

Detailed Description Text (19):
As shown in FIGS. 2, 3, 5, and 6 for example, the other component of the hand-detachable connection includes an elongated coupling 118 that is secured at one end to the air entrance opening 52 of the sack and extends outwardly from the sack. The coupling has an axial opening 120 defined therethrough to permit air to pass through same and between the interior and exterior of the sack. The exterior of coupling 118 is configured to be received within the interior of the connection fitting. The exterior of the coupling has a groove 122 therearound that is configured to seat around and seal against the deformable O-ring 114 of the connection fitting 108 therein when the coupling is inserted into the connection fitting in airtight engagement with the fitting. Groove 122 provides a locking detent to mechanically lock and seal O-ring 114 therein.

Detailed Description Text (35):
As shown in FIG. 16 for example, the control panel of the present invention has a button for SEAT DEFLATE. When the operator presses the SEAT DEFLATE button, the microprocessor activates the two pressure control valves which control the pressure in the sacks supporting the seat zone (Zone III shown in FIGS. 12 and 13 for example) of the support system. The microprocessor signals the pressure control valves controlling the seat zone to align their pistons dump passages with the dump holes in the valve housings in order to permit all of the air in the sacks in the seat zone to escape to the atmosphere through the dump holes. As shown in FIG. 8 for example, when the valve pistons are aligned in this manner, the valve inlets are blocked by the pistons and thus prevented from communicating with the valve passages and valve outlets.

Detailed Description Text (38):
As shown in FIGS. 12 and 13 for example, five pressure zones or body zones preferably include a head zone (Zone 1 or I), a chest zone (Zone 2 or II), a seat zone (Zone 3 or III), a thigh zone (Zone 4 or IV), and a leg and foot zone (Zone 5 or V). Each body zone is supplied with pressurized air from the blower via two separate pressure control valves. In one configuration of the air flow path from the blower to the sacks, one of the pressure control valves controls air supplied to the chambers of each sack on one side of the patient support system for each body zone, and the other pressure control valve controls the air to the chambers on the side of eachsack on the opposite side of the patient support system. In yet another configuration of the air flow path from the blower to the sacks, one of the pressure control valves controls the air supplied to all of the chambers of every alternate sack in a body zone, and the other pressure control valve controls the air supplied to all of the chambers in the remaining alternate sacks in the body zone.

Detailed Description Text (47):
As shown in FIG. 13 for example, each flow diverter valve preferably is mounted within a modular support member 68, and more than one diverter valve 220 can be mounted in a modular support member such as the seat sack support member 94. However, other sack support members 68, such as the head sack support member shown in FIG. 13 for example, may lack a diverter valve. Each diverter valve preferably is mounted between the top and bottom surfaces of each plate 70. As shown schematically in FIG. 11 for example, each diverter valve has a first flow pathway 222 with a first inlet 224 at one end and a first outlet 226 at the opposite end. Each diverter valve further includes a second flow pathway 228

with a second inlet 230 at one end and a second outlet 232 at the opposite end. The flow pathways are mounted and fixed on a rotating disk 234, also referred to as a switching disk 234, that rotates about a central pivot 236.

Detailed Description Text (53):
Moreover, as shown schematically in FIG. 12 for example, the angle of elevation of the head and chest section of the patient support is monitored by an elevation sensing device 242, which sends signals to the circuit board 150 of the modular valve mounting manifold 128. FIG. 12 illustrates electrical signaling pathways by dashed lines and pneumatic pathways by solid lines. The arrows at the ends of the dotted lines indicate the direction of the electrical signals along the electrical pathways. The elevation sensing device detects the angle at which the head and chest section has been positioned, and supplies a corresponding signal to the microprocessor via circuit board 150. Examples of suitable elevation sensing devices are disclosed in U.S. Pat. Nos. 4,745,647 and 4,768,249, which patents are hereby incorporated in their entireties herein by reference. If this elevation information from the sensing device 242 indicates that the angle of articulation exceeds 30.degree., the microprocessor configures the pressure profile to a standard mode of operation and thus cancels any rotation or pulsation that may have been selected by the operator. The rotation mode is cancelled to avoid torquing the patient's body. The pulsation mode is cancelled because the elevation of the patient above 30.degree. reduces the ability to float the patient in the sacks in the seat zone during pulsation of the three sacks therein. Thus, the "bottoming" of the patient during pulsation at elevation angles above 30.degree. is avoided. Upon reduction of the articulated angle below 30.degree., the microprocessor does not automatically resume either pulsation or rotation but requires any mode other than the standard mode to be reset.

Detailed Description Text (55):
Microprocessor 160 has a blower control algorithm which enables microprocessor 160 to calculate a desired reference pressure for the blower. The blower control algorithm preferably calculates this blower reference pressure to be 3 to 4 inches of standard water higher than the highest pressure in the air sacks. Typically, the seat zone (Zone III) has this highest pressure for a given height and weight setting (provided by the operator to the microprocessor) regardless of the elevation of the head and chest sections and whether the patient is lying on his/her side or back. However, a patient with abnormal body mass distribution (which could be caused by a cast for example) may require the highest sack pressure in one of the other zones. If Zone III has the highest sack pressure, as the elevation angle increases, the sack pressure in Zone III increases, and the reference pressure for the blower also increases to equal 3 to 4 inches of standard water above the pressure of the sacks in Zone III.

Detailed Description Text (102):
It is a desirable feature of the vibrational therapy device according to the present invention that vibrating means A, for example, pneumatic vibrating system B, be separately actuable and controllable in any of the modes of operation of the patient support system. If vibratory patient support system 300 of the present invention includes the low air loss patient support system described earlier, it is desired to be able to actuate and control vibrating means A so as not to interfere with or affect the operational mode of the low air loss system. For example, it may desired to actuate vibrating means A while the low air loss patient support system is simultaneously rotating the patient from side to side. Additionally, it may only be necessary to actuate vibrating means A for only brief periods of time at preselected intervals. In this manner, it is desired to have a timing control circuit, within microprocessor 160 for example, for establishing and controlling the period of operation of vibrating means A regardless of the mode of operation of the low air loss patient support configuration.

CLAIMS:

4. The apparatus of claim 3, further comprising a pressure transducer coupled to said controller and in fluid communication with said valve, said pressure transducer operable to provide said controller with second signals that are indicative of air pressure applied to said valve outlet, wherein said controller is operable to receive said second signals that are indicative of air pressure and adjust said first signals sent to said motor based upon said second signals received from said pressure transducer.

8. The apparatus of claim 7, further comprising a pressure transducer coupled to said controller and in fluid communication with said valve, said pressure transducer operable to provide said controller with second signals that are indicative of air pressure applied to said valve outlet, wherein said controller is operable to receive said second signal that are indicative of air pressure and adjust said first signals sent to said motor based upon said second signals received from said pressure transducer.

11. The apparatus of claim 10, further comprising a pressure transducer coupled to said controller and in fluid communication with said valve, said pressure transducer operable to provide said controller with second signals that are indicative of air pressure applied to said valve outlet, wherein said controller is operable to receive said second signal that are indicative of air pressure and adjust said first signals sent to said motor based upon said second signals received from said pressure transducer.

18. The apparatus of claim 17, further comprising a pressure transducer coupled to said controller and in fluid communication with said valve, said pressure transducer operable to provide said controller with second signals that are indicative of air pressure applied to said valve outlet, where in said controller is operable to receive said second signals that are indicative of air pressure and adjust said first signals sent to said motor based upon said second signals received from said pressure transducer.

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TITLE: Fluid powered control system and fail-safe valving system for a fluid powered system

Abstract Text (1):
In a fluid powered system having first and second portions, each normally receiving a supply of fluid under pressure, and a third portion, a fail-safe valving system is fluidly connected in series between the second and third portions. In one embodiment, the fluid powered system is a fluid-powered control system having a plurality of fluid amplifiers whose summed outputs provide an input signal for a flow control valve fluidly and operatively connected to an actuator which is drivingly connected to a movable structure, and the fail-safe valving system is fluidly connected in series between the flow control valve and the actuator for controlling fluid flow therebetween. The actuator is operable to translate the movable structure within a permitted range of movement. The fail-safe valving system has a valving mechanism operable for restricting but not entirely stopping fluid flow between the flow control valve and the actuator upon the occurrence of failure of each fluid amplifier, the restricted flow permitting dampened movement of the movable structure. The fail-safe valving system also has valving mechanism operable to shut off fluid flow between the flow control valve and the actuator upon the occurrence of a loss of supply pressure within the control system. In one embodiment, a position feedback linkage system is operably connected between the movable structure and a movable valving element of the flow control valve for imparting an error signal to the flow control valve, when the fail-safe valving system has been actuated to restrict fluid flow between the flow control valve and the actuator, which error signal causes the flow control valve to operate the actuator to drive the movable structure toward a preselected position within its positional range.

Brief Summary Text (2):
In fluid-powered servo control systems of the type described in the above-referenced application, wherein multiple fluid amplifiers provide an input signal to a flow control valve which positions a movable structure, a dangerous condition may arise if the hydraulic amplifiers should become inoperative, whereupon no position command signal is transmitted to the actuator and whereupon the position of the valving element of the fluid flow valve may become uncontrolled. For example, in such control systems employed in modern aircraft for positioning the movable, airflow control surface elements, such as vertical and horizontal stabilizers, actuator control of a rudder stabilizer may become suddenly lost following actuator failure, and air flow pressures upon the stabilizer may, if not opposed, then cause a sudden, sharp movement of the stabilizer from its previous position, thus producing a major alteration of the aircraft flight characteristics and possibly resulting in complete loss of control by the pilot. When the control system is employed in an aircraft for positioning an aircraft control surface such as a rudder, it is desirable to permit the rudder to return to its centered position, in the event of a partial malfunction of the control system, so that the pilot may retain some degree of control of the craft by the use of the remaining control surfaces. However, it is also necessary that any corrective or fail-safe movement of the control surface element be dampened to prevent undesirably sudden or severe changes

in flight characteristics of the aircraft. It is also desirable that provision be made for preventing complete loss of control of the aircraft in the event of a total failure of the control system as would be caused by a loss of all fluid pressure within the control system. These hazards, and the utility of the fail-safe linkage of the present invention in minimizing them, will be more fully understood from the detailed description to follow. For clarity, and for completeness of disclosure, the multiple channel, servo control system, with its multiple, pressure-responsive monitor valves, will first be described in detail, after which the construction and operation of the fail-safe valving system and the position feedback linkage system will be described with respect to its use in combination with the servo control system.

Brief Summary Text (3):
Fluid-powered control system having multiple, redundant control channels have been proposed for various applications in which reliability is of great importance, such as avionic systems including "fly-by-wire" servo control actuator systems. It is desirable in such systems to eliminate lengthy mechanical linkages, push rods, and the like extending between a control station and a remotely controlled actuator system because such linkages have a degree of flexibility which may distort a given input signal, and because their mass may become a limiting factor when rapidly changing signals are to be transmitted. While present aircraft are designed in such a manner that they normally tend to remain relatively stable in flight, proposed aircraft, known in the art as "control configured vehicles," incorporate certain aerodynamic configurations which, while providing improvements in performance and operating efficiency, result in a decrease in inherent flight stability; they therefore require control systems having extremely rapid response times not practicably obtainable with mechanical control linkages. Thus, the use of fly-by-wire systems has been proposed wherein electromechanical "command" transducers are controlled automatically or by an operator for generating electrical command signals which are transmitted to remote, servo control systems by means of wires rather than mechanical linkages. The servo control systems employ input transducers to convert the electrical control signals into corresponding mechanical or fluidic signals, which may be amplified and then employed, for example, to effect a commanded translation of a movable element such as an aircraft control surface structure. Electrical transducers, sensors, and the like required in such servo systems, however, are susceptible to malfunctions and failure from various causes and often do not have the very high level of reliability required in the control systems of aircraft or space vehicles. To compensate for the unacceptable levels or reliability of such electrical components, as well as that of some non-electrical components, redundant control channels are employed, and various techniques of "majority voting" of multiple components have been devised wherein failed or inoperative control channels are outvoted or overpowered by the remaining channels.

Brief Summary Text (4):
Integration of the outputs of a plurality of redundant channels in a control system may be accomplished by summing the signals of each channel. Conventionally, a plurality of electrical cables from a command station transmit to a servo system a corresponding plurality of redundant, electrical command signals which, as has been suggested, may correspond to the desired position to which a movable element, e.g., an aircraft control surface element, is to be moved. As will be described more fully in the detailed description of the invention, each command signal may then be amplified within the control system, correlated with a position feedback signal corresponding to the current position of the movable element, and the appropriate corrective signal then applied to a respective one of a plurality of signal amplifiers. These amplifiers are typically of the electrohydraulic transducer type producing fluid output signals through two outlets at pressures and/or flow rates which may vary differentially with respect to each other in response to the respective electrical input signal. It is at this stage that the redundant signals are normally integrated and at which majority voting is accomplished. The hydraulic output signals of the transducers may be "summed" by employing a movable piston structure having oppositely directioned piston face areas, the output portions of the multiple transducers having their respective fluid outlets

interconnected with the outlets of like sense of the other transducers, the interconnected outlets of one sense having communication with a respective piston face on the movable structure and the interconnected outlets of the opposite sense having communication with a corresponding but oppositely directional piston face. The electrohydraulic transducers may be of the well-known, flapper type, in which case their nozzles are connected in parallel with each other and with the two piston face areas of the summing structure effectively averaging the differential pressures and flows across the fluid outlets of the transducers. Or, other types of fluid amplifiers may be employed in parallel. In all such cases wherein there is fluid communication between the outlets of the fluid amplifiers, however, there exists the disadvantage that should a leak, or a flow stoppage or restriction, occur in any of the conduits or passageways through which fluid pressure is communicated from the amplifiers to the movable summing structure, the output of the entire system is directly affected and the system may malfunction or become inoperative. Such problems are minimized by a second summing method, which will be termed herein the "force summing method," and in which each of the amplifier fluid outlets of a given sense has communication with a respective one of a plurality of piston face areas formed on the movable summing structure and facing in a first direction, and each of the fluid outlets of the opposite sense has communication with one of a second plurality of piston face areas facing in a second, opposite direction. Fluid amplifiers of the type known as jet pipe, electrohydraulic valves may be employed in such systems and incorporate a fluid supply of constant flow rate which is ejected through a jet nozzle movably mounted in a manner which permits the ejected flow to be directed into either of two outlets or to be proportionally divided therebetween. In any of the systems, a feedback system, such as a mechanical spring element, is preferably connected between each respective fluid amplifier movable element and the summing structure. The feedback elements, as will be more fully described hereinbelow, serve to minimize the output of any amplifier which is in substantial disagreement with the signals of the remaining channels by averaging the signals of all the channels.

Brief Summary Text (6):
Because of this limitation with respect to system response after multiple failures, it may be necessary, in systems in which a high degree of reliability is required, to incorporate monitoring and valving mechanisms operable to sense an abnormal condition in any of the channels and to deactivate or isolate an abnormal channel from the remaining channels. Some relatively complex systems employ sensing and valving devices operative to compare the output of each of a plurality of redundant components, such as transducers or amplifiers, with that of the remaining such components. It can thus be seen that in a system employing a given number n of channels, a minimum number of comparators given by the series [(n-1) + (n-2) . . . (n-n)] are needed for such an approach. Moreover, if it is desired to completely isolate the fluid output signal of the failed channel from the remaining channels interconnected therewith, an undesirably large number of valves, corresponding to the number of passageways interconnected between the several amplifiers, must be provided, or the fluid supplied to each amplifier must pass through n- 1 shut off valves prior to entering that amplifier. A further problem with some such systems is that inoperative fluid amplifiers may impart an undesirably great load or drag opposing any movement of the movable member. This occurs because the fluid pressure control segment of some such amplifiers incorporate restrictive orifices or nozzles through which fluid must pass when the movable summing structure is moved. Furthermore, in those redundant channel control systems employing multiple electrohydraulic transducers whose outputs are interconnected with those of other transducers, monitoring of the fluid pressures at the output stages of the individual transducers is made difficult or impossible because a change in pressure at one transducer tends to distort the fluid output of the other transducers.

Brief Summary Text (11):
Another object is to provide such a control system in which the flow control valve is fluidly connected to drive an actuator and is operable to control the velocity of movement of a load driven by the actuator.

Brief Summary Text (13):
A further object is to provide such a system also having a feedback linkage system operatively connected between the load and the valve element of the flow control valve for imparting an error signal to the flow control valve, when the valving system has been actuated to restrict fluid flow between the flow control valve and the actuator, which error signal causes the flow control valve to operate the actuator to drive the load toward a preselected position within its positional range.

Detailed Description Text (1):
With initial reference to FIG. 1, the control system, identified herein with reference to the system housing 26 to be described, is electrically connected to a remote command station, not shown, via multiple input cables 10, 10A, 10B, and 10C. For illustrative purposes, the control system (26) will be described with respect to its application in an aircraft control system of the "fly-by-wire" type wherein the command station to which redundant input cables 10, 10A, 10B, and 10C extend includes an electromechanical transducer continuously positioned or controlled by the pilot or by an automated, flight control system. The control system (26) of the present invention is employed as a servo system having a duplex hydraulic actuator 11 (FIG. 3), having dual pistons 12 and 13 connected by a piston rod 14 for positioning a load, such as an airfoil 241 (FIG. 7, to be described) or other movable element. In such an application, the actuator piston rod 14 is drivingly connected to the airfoil 241 suitably by means of a linkage 240 (FIG. 7), as will be described hereinafter, attached to a fastening lug 15 mounted on the projecting, distal end of the piston rod 14.

Detailed Description Text (2):
An electromechanical, position sensor system 16 is mounted on the actuator 11 and has axially movable connecting rods 17 extending from the sensor in parallel alignment with the actuator piston rod 14, the distal ends of the connecting rods 17 suitably being fastened to the connecting lug 15 of the piston rod 14 whereby the connecting rods 17 are axially movable in unison with the piston rod 14. The construction and operation of such electromechanical position sensors is well known, and they essentially comprise electromagnetic transducers operable to produce an electrical signal proportional to the position of a movable element. In the present embodiment, four such signals are required, and a sensor system of the linear variable differential transformer type having four sensor elements similar to that available from the G. L. Collins Corp. under part no. LMT199V15 is suitable. Such a sensor system 16 includes four transducer elements, not shown, for producing redundant position signals, first, second, third and fourth position feedback wires 18, 18A, 18B, and 18C being respectively connected to the transducer elements. First, second, third, and fourth difference signal amplifiers 19, 19A, 19B, and 19C, of a generally known type, are connected to the first, second, third, and fourth position feedback wires 18, 18A, 18B, and 18C, respectively, and to the first, second, third, and fourth input cables 10, 10A, 10B, and 10C, respectively. According to principles known to those in the art, the difference signal amplifiers are each operable to compare the command signal received through the respective, associated input cable 10, 10A, 10B, or 10C with the position signal received through the corresponding feedback wire 18, 18A, 18B, or 18C, respectively. Each difference signal amplifier 19, 19A, 19B, and 19C is operable to produce an electrical "error signal" which is proportional to the difference between the commanded, position signal received through the respective, associated input cable 10, 10A, 10B, or 10C and the position signal received through the corresponding, respective, feedback wire 18, 18A, 18B, or 18C, according to methods generally known in the art of servo control mechanisms. The difference signal amplifiers 19, 19A, 19B, and 19C amplify the difference signals somewhat, and additional amplifying circuits (not shown) may also be provided in series therewith.

Detailed Description Text (13):
The feedback elements 77, 77A, 77B, 77C are suitably elongated leaf spring elements, extending toward

the adjacent annular grooves 76 of the respective, adjacent piston structures 73, 73A, 73B, 73C when both the jet nozzles 22 and the elongated structure 37 are positioned centrally within their respective ranges of movement as shown in the drawing and as described more fully hereinbelow, and each feedback element is provided on is distal end with a respective, spherical bearing member 79 which is arranged to seat within the annular groove 76 of the respective, adjacent piston structure 73, 73A, 73B, and 73C and to have sliding contact therein with the pistons 74, 75. Communication is provided, through suitable passageways within the first housing segment 27, between the annular grooves 76 of the first elongated member 38 and the first fluid return passageway 69 and, within the second housing segment 28, between the corresponding annular grooves of the second elongated structure 39 and the second return passageway 70.

Detailed Description Text (16):
The first and second chamber portions 82, 83 adjacent each piston structure 73, 73A, 73B, and 73C are of sufficient width and are appropriately spaced to permit a desired degree of axial movement of the piston structures, and thus of the movable structure 37, as determined by the flow characteristics of the flow control valve 53 and the desired maximum actuator velocity.

Detailed Description Text (18):
With additional reference now to FIG. 3, the flow control valve 53, the valve element 52 of which is operatively connected to the movable structure 37 (FIG. 1) by means of the first and second rocker arms 46, 47, is of the known type wherein a movable valve element (52) controls the division of fluid flow from the valve into two outlets. The flow control valve 53 employs a substantially tubular housing 90 which is sealingly mounted within a corresponding cavity within the housing 26. More precisely, the valve housing 90 extends equidistantly within the first and second housing segments 27 and 28 along an axis which is parallel to the longitudinal axis of the summing structure 37 (FIG. 1). With respect to the portion of the flow control valve 53 extending within the first housing segment 27, which portion constitutes a single, flow control valve redundant to and substantially identical to the valve portion extending within the second housing segment 28, there is provided through the valve housing 90 a fluid inlet 91 having communication with the first source of fluid under pressure P.sub.1 and first and second fluid outlets 92, 93 which communicate, through conduits 94, 95, respectively, with the interior of the portion of the actuator 11 containing the first piston 12 at respective locations spaced on opposite sides of the first actuator piston 12, conduits 94 and 95 communicating through portions of the valving system (170), to be described. The first and second valve outlets 92, 93 are spaced in the first and second directions, respectively, along the length of the valve housing 90 (leftwardly and rightwardly, as viewed in the drawing) from the fluid inlet 91. First and second fluid return passages or orifices 96, 97 are also formed through the valve housing 90, the return passages 96, 97 both having communication with the first fluid return outlet R.sub.1. The first return orifice 96 is spaced, in the first direction, beyond the first fluid outlet 92, and the second return orifice 97 is spaced, in the second direction, beyond the second fluid outlet 93.

Detailed Description Text (19):
The valve element 52 is sealingly and slidingly associated along its length with the valve housing 90 but is reduced in diameter along portions of its length which may become adjacent the fluid inlet 91, the outlets 92, 93, and the return orifices 96, 97, during any permitted position of the valve element 52. First and second, mutually spaced, annular lands 98, 99 extend circumferentially around the valve element 52, the first land 98 being positioned between the first return orifice 96 and the inlet 91, and the second land 99 being positioned between the inlet 91 and the second return orifice 97. Upon the movable summing structure 37 (FIG. 1) being positioned in its central position as shown in FIG. 1, the first and second lands 98 and 99 of the valve element 52 are in register with the first and second fluid outlets 92 and 93, respectively, the width of the lands 98, 99 being substantially the same as the diameter of the outlets 92,

93, respectively. For convenience of assembly, first, second, third fourth, and fifth external annuli 103, 103A, 103B, 103C, and 103D are formed circumferentially of the valve housing 90, respectively adjacent and in communication with the first return orifice 96, the first fluid outlet 92, the fluid inlet 91, the second fluid outlet 93, and the second return orifice 97. Thus, the external annuli 103, 103A, 103B, 103C, and 103D permit communication between the conduits 96, 92, 91, 93 and 97 and the respective associated passageways, e.g., 71, 94, 29, 95, without precise orientation of the valve housing 90, upon its longitudinal axis, within the housing 26. The portion of the valve 53 extending within the second housing segment 28 is similarly constructed, and has corresponding outlet conduits 100, 101 communicating, through portions of the valving system 170, with the interior of the actuator 11 on opposite sides of the second piston 13.

Detailed Description Text (20):
From the above description it will be recognized by those in the art that the flow control valve 53 is of a type operable, in response to axial movement of the valve element 52, to control the rate and direction of flow of fluid under pressure through conduits 94 and 95, and through conduits 100, 101, to the actuator 11, thereby controlling the direction and velocity of movement of the duplex piston 14.

Detailed Description Text (21):
With respect now to the operation of the portions of the control system 26 thus far described, as has been previously suggested, the operation of servo system components such as the fluid amplifiers 19, 19A, 19B, 19C and the corresponding transducers of the position sensor system 16 (FIG. 3) in response to electrical command signals received from a remote control station is generally known in the art. Summarily, and with respect to the first channel, an electrical signal of a current level which may vary over a predetermined range is supplied to the difference signal amplifier 19 through cable 10. The signal supplied at any given moment corresponds to a position, within a corresponding range of physical movement, at which it is desired to position the actuator piston 14 and its load, which in the case of the present embodiment and as has been previously stated, may be a movable airfoil or control surface element 241 (FIG. 7). An electrical signal corresponding to the current position of the piston 14 is produced by an element of the position sensor 16, suitably by an inductive modification of an externally supplied signal, as occurs in the present, commercially supplied LVDT sensor 16 previously named, and conducted by feedback wire 18 to the difference signal amplifier 19. The difference signal amplifier 19 compares the feedback signal received from the sensor system 16 with the command signal received via input cable 10 and, if the piston rod 14 is not in the commanded position, difference signal amplifier 19 produces an error signal, or modified command signal, which it electrically amplifies and which is then conducted via output lead 20 to the first fluid amplifier 21. The fluid amplifier 21 is operative, as was previously discussed, to translate the modified command signal received through lead 20 into a corresponding differential pressure signal across its outlets 35, 36 (absent any movement of the summing structure 37) and thus, across the first and second piston face areas 80, 81 of the first piston structure 73. While the operation of only the servo components associated with the first fluid amplifier 21 have been described, it will be understood that the corresponding components associated with the second, third, and fourth fluid amplifiers 21A, 21B, and 21C function similarly to provide redundant, modified command signals, to the second, third, and fourth amplifiers.

Detailed Description Text (22):
It will be recalled from the initial discussion of prior-art control systems that it is the electrical components of such systems which are the most susceptible to failures or malfunctions and for which the greatest degree of redundancy must therefore be provided. Thus, a redundancy level of four is incorporated into the system 26 with respect to the fluid amplifiers 21, 21A, 21B, 21C, the difference signal amplifiers 19, 19A, 19B, 19C, and the four-element position sensor system 16, whereas only two operating sections are required for the duplex actuator 11 and for the flow control valve 53, respectively.

Assuming that each element of the position sensor system 16, and each of the difference signal amplifiers 19, 19A, 19B, 19C and the fluid amplifiers 21, 21A, 21B, and 21C are functioning properly and that identical, redundant command signals are received by difference signal amplifiers 19, 19A, 19B, and 19C, then the differential fluid pressures produced by each fluid amplifier 21, 21A, 21B, 21C are substantially identical, and each imparts a force upon the summing structure tending to urge it in the same direction, if the commanded position differs from the current position of the actuator piston 14.

Detailed Description Text (23):
Upon the amplifier 21 receiving an electrical command signal moving its nozzle 22 in the first direction, for example, the feedback element 77 flexes and imparts an oppositely directioned torque on the nozzle 22 which is proportional to the current level of the electrical signal. Absent the feedback elements 77, 77A, 77B, 77C, since there is relatively little resistance to axial movement of the summing structure 37 and the valve element 52, relatively minor electrical signals to the amplifiers 21, 21A, 21B, 21C would tend to move the nozzle 22 sharply to the extreme rightward or leftward positions, thereby causing sharp, severe movement of the summing structure 37 and valve element 52 which could cause damaging transient pressures within the hydraulic conduits 94, 95, 100, 101 leading to the actuator 11 and/or dangerously severe accelerations of the actuator piston 14 and its load.

Detailed Description Text (49):
The valve 33 thus remains closed so long as fluid under pressure is received through the first port 149. Upon the excessive differential pressure between chamber portions 82, 83 (FIG. 1) being removed, however, the valve 33 may be reopened by temporarily shutting off fluid flow through the port 149, whereupon the second coil spring 146 acts to move the valve member 131 back to its centered position shown in FIG. 4. Such temporary termination of fluid flow to the valve 33 may be accomplished by actuation of the first remotely controllable shutoff valve 31 (FIG. 2). A normally open, pressure sensing switch 102 (FIG. 1) may be connected to the supply conduit 32 between the valve 33 and the fluid amplifier 21 for closing a circuit to a remote warning light (not shown) when the fluid pressure within the supply conduit 32 is shut off. As will be apparent to those in the art, operation of the monitor valve 33 to shut off fluid flow upon the occurrence of a differential pressure from a loss of pressure in the first chamber portion 82 (FIG. 1) with respect to that in the second chamber portion 83 causes the valve member 131 to move to the left in the second chamber portion 83 causes the valve member 131 to move to the left from its third positional range, through its second and into its first range, in the same manner as has been described above with respect to rightward movement. Upon the valve member 131 entering its second positional range, fluid within the first, annular chamber portion 135 is permitted to drain through the first axial bore 159, the passageway 164, and the drainage opening 155.

Detailed Description Text (51):
The first and second chamber portions 135, 137, in cooperation with the valve first and second, annular piston face areas 136, 138 and the valve housing 111 and in cooperation with the axial bores 159, 160, the annuli 156, 157, 158, and the restricted passageways 161, 162, 163, 164, provide a latching means, employing fluid pressure, for moving the valve member 131 entering its second positional range when fluid under pressure is received through the first port 149 and, alternatively, for moving the valve member from its fourth to its fifth positional range upon the valve member entering its fourth positional range. The latching means additionally comprises means, actuated by fluid under pressure received through the first port 149, for constraining or "latching" the valve member 131 in its first positional range upon the valve member being moved to its first positional range by the latching means; and, alternatively, for constraining the valve member 131 in its fifth positional range upon the valve member being moved to its fifth positional range by the latching means.

Detailed Description Text (55):

From the foregoing description of the servo control system 26, the interrelationship of the fail-safe valving system 170 with portions of the servo control system can be more clearly discussed; the failsafe valving system 170, termed hereinafter the valving system 170, will now be described in detail. The valving system 170, in its preferred embodiment, includes first and second, elongated valving members 171, 172 slideably and sealingly inserted within corresponding, first and second, cylindrical valving chambers 171', 172' extending within the first and second housing segments 27, 28, respectively, along respective axes suitably parallel to the longitudinal axis of the flow control valve 53 but spaced laterally from each other. The outboard end portions of the valving members 171, 172 have respective, coaxially, and outwardly projecting plungers 173, 174 of reduced diameters and which extend toward and, upon the first and second valving members 171, 172 being positioned in their extreme outboard positions (their extreme leftward and rightward positions, respectively, as viewed in the drawing), butt against first and second elongated stop members 175, 176, respectively, which are slideably mounted within the outboard end portions of the valving chambers 171', 172'. The first and second, elongated stop members 175, 176 have respective head portions 177, 178 adapted to seat against first and second valving member plug members 180, 181, respectively, which are suitably threadingly engaged within the parallel valving chambers 171', 172', respectively, and positioned adjacent the external, side surfaces of the first and second housing segments 27, 28 for facilitating assembly and maintenance of the valving system 170. The first elongated stop member 175 and the plunger 173 of the first valving member 171 extend coaxially within a first, coiled, actuating spring 182 footed under compression between and against the first stop member head portion 177 and the step defined between the first plunger 173 and the non-reduced, outboard end portion of the first elongated valving member 171, for resiliently urging the valving member 171 inboard, or in the second direction. Similarly, a second actuating spring 183 foots against the second stop member head portion 178 and the second, elongated valving member 172 for urging the second valving member 172 in the first direction.

Detailed Description Text (65):
The first elongated valving member 171 has first and second annular grooves 230, 231 formed circumferentially thereof at locations respectively in register with the conduits 94, 95 communicating with the first and second fluid outlets 92, 93, respectively, of the flow control valve 53 upon the valving member 171 being in its fully retracted position as in FIG. 3. The conduits 94 and 95 have portions 94A and 94B, respectively, communicating between the servo valve outlets 92, 93, respectively, and the first valving chamber 171'; upon the first valving member 171 being positioned in its retracted position (abutting the first elongated stop member 175), conduit portions 94A and 95A communicate with the valving member 171 first and second annular grooves 230, 231. Portions 94B, 95B of the conduits 94, 95, respectively, then similarly communicate between the first and second annular grooves 230, 231, respectively, and with the actuator 11 (FIG. 4). Similarly, the second elongated valving member 172 has corresponding, first and second annular grooves 232, 233 which communicate, upon the second elongated valving member 172 being positioned in its retracted position abutting the elongated stop member 176, with corresponding portions 100A, 100B of the conduit 100 and with portions 101A and 101B of conduit 100, portions 100B and 101B communicating with the actuator 11. The width and depth of annular grooves 230, 231, 232, and 233 are sufficiently great to permit substantially unrestricted fluid flow through conduits 94, 95, 100, and 101 via the respective associated grooves. The first valving member 171 additionally has first and second annular orifices 234, 235 respectively spaced, adjacently, in the first direction from the first and second grooves 230, 231 and oriented to communicate, respectively, between conduit portions 94A, 94B and between conduit portions 95A, 95B upon the first valving member 171 being positioned in a medial position in which the first valving member 171, the first end portion 184 of the rocker arm 186, the first stepped plunger member 194, and the stop member 201 are, successively, in mutual contact and in which the stop member head portion 202 seats against the radial step between the first chamber midportion 192A and its second enlarged portion 192C.

Detailed Description Text (67):
The first actuating spring 182 continuously urges the first elongated valving member 171 in the second direction, and the second actuating spring 183 urges the second elongated valving member 172 in the first direction, thus tending to rotate the third rocker arm 186 in a counterclockwise rotational direction as viewed in the drawing. Such counterclockwise rotation of the rocker arm 186 is normally opposed by the stepped plunger members 194, 211 because of fluid pressures which urge the plunger members against the rocker arm 186, thus opposing the action of the actuating springs 182, 183. Fluid under pressure is conducted from the first and second fluid supply conduits 32, 32A (FIG. 1), via the first and second fluid receiving conduits 220, 220A, respectively, to the second elongated chamber 210 at respective, fluidly discrete portions thereof communicating with the second and first piston faces 214, 213 of the second plunger 211, as have been described previously. Similarly, fluid under pressure is conducted from the third and fourth fluid supply conduits 32B, 32C (FIG. 1), via the third and fourth, fluid receiving conduits 220B, 220C, respectively, to portions of the first chamber 192 respectively communicating with the first and second piston faces 206, 207 of the first plunger 194. The spring rates of springs 182, 183 are selected to permit compression of both springs by the force exerted upon any one of the piston faces 206, 207, 213, and 214 by the fluid pressures available at the respective, associated fluid supply conduits 32B, 32C, 32A, and 32 (FIG. 1) during normal operation of the respective fluid amplifiers 21B, 21C, 21A, and 21. That is, the elongated valving members 171, 172 are restrained in their retracted positions by a normal fluid pressure acting upon at least one of the piston faces 206, 207, 213, 214, and the valving system 170 is thus operative to permit unrestricted flow of fluid to the actuator 11 through conduits 94, 95, 100, and 101, via annular grooves 230, 231, 232, and 233 so long as pressure is received at any one of the valving member piston faces 206, 207, 213, 214. Thus, unrestricted communication is permitted between the flow control valve 53 and the actuator 11 as long as supply fluid is present at any one of the fluid amplifiers 21; i.e., as long as any one of the monitor valves 33 remains open to permit fluid flow to the respective, associated, fluid amplifier. As will be discussed in a later section, the valving system 170 has utility, in combination with the servo control system 26, apart from the fail-safe linkage system 58. The valving system 170 is, however, particularly adapted for use in combination with the linkage system 170, as will be understood from the description to follow.

Detailed Description Text (68):
With reference now to FIG. 7, the fail-safe linkage system 58 will now be described. The system housing 26 and the actuator 11 are bolted together, or connected by other suitable means, and are each connected to portions of the fail-safe linkage system 58. In FIG. 7, the housing 26 and actuator 11 are shown in the same scale, in contrast to the diagrammatic representation of FIGS. 1-4 wherein components illustrated in FIGS. 1-3 are enlarged, for clarity, relative to the actuator 11 of FIG. 4.

Detailed Description Text (69):
According to practices generally known in the art, the lug 15 of the actuator piston rod 14 suitably is pivotally connected to a driving arm 240 which extends generally perpendicularly from the piston rod 14 toward the load, which, in the present example, comprises an airfoil 241, such as a horizontal stabilizer, rotatable about an axis extending spanwise and shown diagrammatically in cross-section in FIG. 7. The lug 15 and arm 240 are suitably pivotally connected by an axle or pin 242 extending between spaced clevis segments of the lug 15 and rotatably supporting the arm 240 between the clevis segments. The stabilizer 241 may be of the type pivotally supported upon an axle 243 rotatably journalled within a bearing assembly 244 which is seated upon an axle 243 rotatably journalled within a bearing assembly 244 which is seated within structure 245 rigidly affixed to the frmae of the aircraft as indicated diagrammatically at 246. Thus, the actuator 11 is operable, by generally linear movement of the piston rod 14, to cause rotational movement of the stabilizer 241 within bearing assembly 244, the axes of axles 243 and 242 extending generally parallel to the major plane of the stabilizer 241. The external housing 249 of the actuator 11 is affixed to the aircraft frame by any suitable means permitting freedom of

rotation of the actuator housing 249 in a vertical plane, and is preferably connected by a trunion assembly 250 having a vertical clevis grounded to the aircraft frame as indicated at 251 and connected to a lug 252 formed on the left end of the actuator housing 249 for permitting pivotal or orbital movement of the actuator housing 249 upon the trunion 250.

Detailed Description Text (70):
The linkage elements of the fail-safe linkage system 58 will now be described. For convenience and clarity of description, the actuator 11 and piston rod 14, which extend from the trunion 250 in the first direction toward the lug 15, will be described as extending horizontally, from left to right as they are shown in the drawing, the control system housing 26 being considered as positioned directly above the actuator 11 and the driving arm 240 extending upwardly and in a vertical plane. These positional terms (e.g., leftwardly, rightwardly, upwardly, downwardly) should not, of course, be considered as limiting the system to any particular orientation but are rather for illustrative and descriptive purposes only. It will be initially noted that, in the present embodiment, the movable elements of the linkage system 58 are pivotable upon respective horizontal axes which are all parallel to the rotational axes of the pin 242 and the stabilizer axle 243 just described, and also parallel to the rotational axes of the rocker arms 46, 47, 186 (FIGS. 1 and 3) within the housing 26. The pivotal structures now to be described, e.g., the arm 273, idler arm 280, cam member 292, and crank 303 (all to be described), are pivotable about respective axes extending parallel to the pin 242 and axle 243, and the respective end portions of the movable, pivotal elements comprise clevis and bearing connections, to be described, in which the clevis slots are parallel to a plane extending perpendicularly of the pivotal axes.

Detailed Description Text (72):
With added reference to FIG. 11, a horizontally extending torque tube 263 is affixed to the walking beam structure 262 (in a manner to be more fully described below) at a location spaced from the end portions of the walking beam. The walking beam structure 262 has a first portion 262A welded to and extending generally upwardly from the center of the torque tube 263 to the lug 260, and a second portion 262B extending in the opposite direction from the torque tube 263. A two-member arm structure 264 comprises first and second parallel arms 264A, 264B extending from and fixedly connected to the torque tube 264 on opposite sides of the walking beam first portion 262A. The distal ends of the arms 264A and 264B are positioned on opposite sides of a raised portion 265 of the actuator housing 249 adjacent the upper, right end portion of the housing 249. A bolt 266 extends through the raised housing portion 265 and through respective bearings, not shown, seated in the distal end portions of the arms 264A and 264B for pivotally supporting the arms and the walking beam structure 262 from the actuator housing 249, permitting pivotal movement of the walking beam structure 262 in a vertical plane. The lower, second portion 262B of the walking beam structure 262 is approximately U-shaped in plan, having legs, whose ends are welded or otherwise affixed to opposite end portions of the torque tube 264, which extend downwardly from the torque tube on opposite sides of the actuator housing 249 and which are connected by a curved, approximately semicircular lower portion 267 extending below the actuator housing 249 and spaced sufficiently from the housing 249 to permit a desired degree of pivotal movement of the walking beam structure 262, as will be understood more fully from the description hereinbelow of the operation of the linkage system 58. The curved portion 262, at its lowermost portion, is welded to a link member 268 (FIG. 7) which extends leftwardly and downwardly from its connection to the walking beam structure 262. The distal end portion of the link member 268 is vertically clevised for receiving a lug 270, which is, in turn, attached to the distal ends of the connecting rods 17 of the position sensor system 16. A bolt 271 extends horizontally through the distal end portion of link member 268 and is rotatably journalled within lug 270 for permitting approximately horizontal and linear movement of the connecting rods 17 in response to pivotal movement of the walking beam structure 262. A vertically clevised base portion 272 of the outer housing of position sensor system 16 is pivotally connected to the actuator housing 249 for permitting pivotal movement of the sensor system 16 about a horizontal axis

for permitting the connecting rods 17 to directly follow the pivotal movement of walking beam structure 262. The position sensor system 16 also includes a horizontally extending, tension spring, not shown, continuously urging the connecting rods 17 and lug 270 to the left, thus tending to apply a clockwise torque to the walking beam structure 262 for imparting a load to all the movable elements of the linkage system just described, thus further reducing any non-pivotal relative movement of the movable elements.

Detailed Description Text (73):
When the actuator piston 14 is approximately centered within its range of movement, the walking beam structure 262 is positioned in a medial pivotal position, as viewed in FIG. 7, wherein it extends upwardly and somewhat rightwardly from its connection to link member 268. The arms 264A, 264B of the two-member arm structure 264 then extend leftwardly and downwardly from the torque tube 263. The arm structure 264, permitting pivotal movement of the walking beam structure 262 about the axis through bolt 266, is employed for convenience in centering the longitudinal axes of the connecting rods 17 and of the elongated linkage rod 253 within their respective ranges of pivotal movement relative to walking beam structure 262, according to practices well known in the art, and may therefore be unnecessary in other embodiments of the system wherein, for example, the walking beam structure is pivotal about the longitudinal axis of its torque tube 263.

Detailed Description Text (82):
With primary reference to FIG. 3, the operation of the fail-safe valving system 170 will now be described. The valving system 170 will be initially described without reference to the linkage system 58, i.e., in an application wherein the fail-safe linkage system 58 is omitted (not shown), whereby axle 188 connected to the third rocker arm 186 is free to rotate and is not connected to the crank 303, wherein the remaining elements of the fail-safe linkage system (including the walking beam structure 262, drag link structure 275, and elonated linkage rod 253) are omitted and whereby the position sensor system 16 has its movable connecting rods 17 connected directly to the actuator piston rod lug 15 (as shown in FIG. 4 and described initially with respect to the electromechanical, position feedback arrangement of the servo system 26) rather than to the walking beam structure 262 as shown in FIG. 7. Such an arrangement has utility in applications wherein there is no need to actively move the movable load structure 241 to a particular, predetermined position within its range of movement in the event of a failure of the electrical portions of the servo control system 26, but wherein it is desired to dampen any movement of the load structure subsequent to such a failure.

Detailed Description Text (83):
More specifically, assume that the servo control system 26 is employed to position an aircraft rudder. While the load 241 has been illustrated as a horizontal stabilizer, for purposes of the description to follow of the valving system 170, reference to the load 241, in the immediately following paragraphs, will be to a rudder (241). As has been previously suggested, the springs 182, 183 of the fail-safe valving system 170 (FIG. 3) have respective spring rates chosen to permit the valving members 171, 172 to be restrained in their retracted positions by fluid pressure (received from any one of the supply conduits 32, 32A, 32B, and 32C respectively and discretely communicating with the fluid amplifiers 21, 21A, 21B, and 21C) acting upon any one of the piston faces 206, 207, 213, or 214, whereupon the valving members 171, 172, maintained in their fully retracted positions as shown in FIG. 3, permit substantially unrestricted fluid communication between the flow control valve 53 and the actuator 11 through the annular grooves 230, 231, 232, and 233 then positioned in register with the conduits 94, 95, 100, and 101. Assume now that a failure of the control system 26 occurs wherein electrical malfunctions occur causing failure of all the fluid amplifiers 21, 21A, 21B, and 21C whereby all the monitor valves 33, 33A, 33B, and 33C are actuated to shut off the supply flow to each of the fluid amplifiers and wherein the pressure of any supply flow to the flow control valve 53 is reduced, e.g., by a pressure reducing valve,

not shown, actuated upon failure of the fluid amplifiers, and connected to reduce pressure within the fluid supplies 29, 30. (Alternatively, such reduced pressure within the fluid supplies 29, 30 could result, in military use, when enemy fire resulted in damage to and leakage of fluid from, the fluid supplies whereby their pressure levels are reduced). The springs 182, 183 of the valving system 170 then are unopposed by fluid pressures acting upon the plunger member piston face areas 206, 207, 213, 214, and are thus free to move the elongated valving members 171, 172 to their partially projecting, medial positions, previously described, in which the restrictive orifices 234, 235, 236 and 237 are in register with the conduits 94, 95, 100, and 101, respectively, thus causing the third rocker arm 186 to rotate in a counterclockwise rotational direction and causing the plunger members 194, 211, to be moved into contact with the stop members 201, 212. Fluid which is ejected from the elongated chambers 192, 210 by this inward movement of the plunger members 194, 211 passes through the fluid receiving conduits 220, 220A, 220B, and 220C, the fluid supply conduits 32, 32A, 32B, and 32C (FIG. 1), respectively, and then through the fluid amplifier fluid inlets 34 to the fluid amplifiers 21, 21A, 21B, and 21C, from which drainage is permitted through the drainage passageways 69, 70. While the valving members 171, 172 are maintained in this medial position, the restrictive orifices 234, 235, 236, and 237 thus permit restricted communication between the flow control valve 53 and the actuator 11, which prevents sudden or rapid, uncontrolled movement of the actuator pistons 12, 13 (FIG. 4) and of the movable load structure 241.

Detailed Description Text (84):
Absent the valving system 170 connected between the flow control valve 53 and the actuator 11, there would be no assurance that such uncontrolled movement would not occur. That is, at the moment of failure of the fluid amplifiers 21, 21A, 21B, and 21C, it is improbable that the flow control valve element 52 will be precisely centered in the "null" position, shown in FIG. 3, wherein the valve outlets (e.g., 92, 93) communicating with the fluid conduits 94, 95, 100, and 101 are closed by the valve member lands (e.g., the first and second lands 98, 99). Thus, the lands (e.g., 98, 99) will in all probability be positioned at least slightly out of register with the corresponding valve outlets, whereupon fluid flow is permitted through the flow control valve 53 to or from the drainage outlets 71, 72 and the sources of fluid under (not reduced) pressure 29, 30, whereupon there is no substantial resistance to movement of the actuator pistons 12, 13 caused by restriction of fluid flow between the actuator 11 and the flow control valve 53. With the valving members 171, 172 positioned in their medial positions as described above, however, any fluid flow between the flow control valve 53 and the actuator 11 is restricted, whereby any subsequent movement of the actuator piston rod 14 and the load structure 241 is damped, and whereby sudden, rapid movement of the load structure is prevented. When the actuator 11 is drivingly connected to a load structure 241, such as a rudder, which tends to move to an acceptable or safe portion of its positional range during normal operation (e.g., to an approximately centered position), the valving system 170, when in its medial positional mode, permits restricted or dampened movement of the load to its acceptable range while preventing violent, uncontrolled movement of the load in either direction. As will be understood by those in the art, the size of the restrictive orifices 234, 235, 236, and 237 may be adjusted to permit a desired degree of restriction of fluid flow, and thus, a desired degree of damping of load structure movement, appropriate to any of a variety of applications.

Detailed Description Text (85):
Further counterclockwise movement of the third rocker arm 186 and further resultant, outward movement of the plunger members 194, 211 is normally prevented by the stop members 201 and 212 because of fluid pressures acting upon the outwardly facing piston face areas defined by the stop member head portions 202, 216, respectively. That is, fluid under pressure is normally received within the third portion 192C of the first elongated chamber 192 via the inlet passageway 225 which has communication with the second source 30 of fluid under pressure via outlet orifice 228 of the flow control valve 53. Similarly, fluid under pressure received from the first fluid source 29 is normally conducted to the outboard piston face of the second stop member 212 through the first inlet passageway 222. The stop

member head portions 202, 216 are of sufficient area, relative to the pressures received, to prevent movement of the stop members 201, 212 from their projected positions, shown in FIG. 3 and described previously with respect to the medial position of the valving members 171, 172, by the action of springs 182, 183 so long as fluid under pressure is applied against either the first stop member 201 from the second fluid source 30 or against the second stop member 212 from the first fluid source 29. Upon the occurrence of a total failure of both fluid sources 29, 30, wherein all supply fluid pressure is lost, however, pressure induced forces opposing movement of the stop members 201, 212 are lost, whereupon the compressed springs 182, 183 act to move the valving members 171, 172 to their fully projected positions in which the third rocker arm 186 is rotated to its extreme counterclockwise position and wherein the stop members 201, 212 are translated outwardly, by outward movement of the plunger members 194, 211, until the stop members 201, 212 abut the chamber transverse end walls 205, 217. In this fully projected position, non-grooved portions of the valving members 171, 172 respectively extending outwardly (relative to the third rocker arm 186) from the first, second, third, and fourth restrictive orifices 234, 235, 236, 237 are in register with the conduits 94, 95, 100, and 101, respectively, and serve to prevent any further substantial fluid flow between the flow control valve 53 and the actuator 11, thus locking the actuator piston rod 14 and the load 241 in their current positions. This locking function is of utility in applications wherein any movement of the load 241 subsequent to a total failure of the fluid supply system 29, 30, whereby no fluid under pressure is supplied to the flow control valve 53, is undesirable or dangerous.

Detailed Description Text (86):
It thus can be understood that the valving member 171, 172, along with the rocker arm 186, comprise a valving structure movable between a first position, in hwich the valving members 171, 172 are retracted within the chambers 171', 172' as seen in FIG. 3, a second (medial) position, and a third, fully projected postiion, as described above. The annular grooves 230, 231, 232, 233 comprise porting means permitting substantially unrestricted fluid flow between the flow control valve 53 and the actuator 11 when the valving structure is in its first position, and the restrictive orifices 234, 235, 236, and 237 comprise orificial means permitting restricted fluid flow between the flow control valve and the actuator upon the valving structure (171, 172, 186) being in its second position. Upon the valving structure being in the third position, wherein the valving members 171, 172 are in their fully projected positions, the non-grooved portions of the valving members which block the conduits 94, 95, 100, 101 thus comprise valving means preventing any substantial fluid flow between the flow control valve 53 and the actuator 11 and preventing any substantial subsequent movement of the movable load structure 241. The plunger members 194, 211, with their associated, stepped, enclosing chambers, comprise pressure responsive means, having fluid communication with the fluid amplifiers 21, 21A, 21B, 21C, urging the valving structure (171, 172, 186) toward its first position and constraining it therein so long as fluid pressure is received through conduits 220, 220A, 220B, or 220C from any one of the fluid supply conduits 32, 32A, 32B, or 32C, i.e., so long as any one of the fluid amplifiers 21, 21A, 21B, 21C is operative, or has not been shut off by an interruption of its supply flow by the respective, associated monitor valve 33, 33A, 33B, or 33C. The springs 182, 183 comprise means urging the valving structure (171, 172, 186) toward its third position and operable to move the valving structure to its second position when the plunger members 194, 211 do not receive fluid pressure from any of the supply conduits 32, 32A, 32B, or 32C. The stop members 201, 212 comprise means normally preventing movement of the valving structure from its second to its third position but permitting movement of the valving structure to its third position upon the occurrence of a loss of fluid pressure upstream of the flow control valve 53 (resulting in a loss of pressure downstream thereof at fluid passageways 222 and 225). The restrictive orifices 234, 235, 236, and 237, in combination with the plunger members 194, 211, the stop members 201 and 212, and the springs 182, 183, comprise means restricting any fluid flow between the flow control valve 53 and the actuator 11 upon the occurrence of a loss of supply flow to each fluid amplifier but permitting a degree of fluid flow permitting dampened movement of the movable structure.

Detailed Description Text (87):
The servo control system 26 may be termed a fluid powered control system of a type having a first portion (e.g., constituting the initial amplification stage provided by the fluid amplifiers 21, 21A, 21B, and 21C); a second portion (comprising the flow control valve 53) to which the first portion is operatively connected by means of the movable summing structure 37 and the first and second rocker arms 46 and 47; and a third portion, constituting the actuator 11, with which the second (flow control valve) portion is fluidly and operatively connected. The fail-safe valving system 170 is thus fluidly connected in series between the second and third, control system portions.

Detailed Description Text (88):
While a preferred embodiment of the fail-safe valving system 170 has been described with reference to its application in a particular control system 26, it will be understood that various modifications of the valving system from the specifically described embodiment may be made as required for other applications. For example, in a less sophisticated servo control system, not shown, employing an actuator having only one piston rather than the duplex piston arrangement 12, 13 of the illustrated actuator 11, a flow control valve having only components corresponding to the flow control valve portions extending within one of the housing segments 27, 28 would be employed, and the flow control valve would thus have only one fluid outlet, e.g., corresponding to the outlet 92, and one fluid return or inlet port corresponding to inlet port 93. In such a simplified control system, there remains no need for fluid isolation of the portions of the valving system 170 respectively mounted within the first and second housing segments 27 and 28. Thus, a simplified embodiment of the valving system, not shown, suitable for such a control system incorporates a single movable valving element, such as the first valving member 171, fixedly connected to or continuous with a stepped plunger member such as the first plunger member 194 (the rocker arm being omitted unless necessary for transmission of positional signals to a position feedback linkage system, as will be described hereinafter). A single stop member such as the first stop member 201 is fluidly connected to the single flow control valve segment, and a single compressed spring (or its functional equivalent) such as the first spring 182 urges the combined movable element toward the stop member.

Detailed Description Text (92):
The drag link structure 275, in use, is adjusted to a length which, upon the (horizontal) stabilizer 241 being in a desired, preselected position within its range of movement and upon the cam member 292 being in its engaged position (FIG. 12), results in the second rocker arm 47 being oriented in a position which axially aligns the valve element 52 of the flow control valve 53 in its null position (FIG. 3), in which no fluid flow is transmitted to the actuator 11. Assume now that the horizontal stablizer 241 is rotated in a counterclockwise direction, as viewed in FIG. 7, to an upwardly inclined position and that the cam 292 is disengaged. This counterclockwise rotation of the load 241 and the driving arm 240 induces a clockwise rotation of the walking beam structure 262, translating the drag link structure 275 to the right and causing rotation of the idler crank 280 in a counterclockwise direction upon its upper bearing 286 (FIG. 6). Assume now that a failure of the servo system occurs in which fluid flow to each of the fluid amplifiers 21, 21A, 21B and 21C is shut off by the monitor valves 33, 33A, 33B, and 33C. Such a failure causes the valving structure (186, 171, 172) of the fail-safe valving system 170 to move to its second position, as has been previously explained, whereupon counterclockwise rotation of the crank 303 (FIG. 12) brings the cam element 292 into contact with the cam follower 290. Because of the previous counterclockwise rotation of the idler crank 280, the cam follower 290 is now positioned somewhat rightwardly of the horizontal center of the concave cam surface of the cam member 292; thus, as the toggle linkage spring 317 urges the cam member 292 into its fully engaged position, the cam follower 290 will be translated leftwardly by the cam 292, i.e., rotated in a counterclockwise direction upon the lug 276 of the drag link structure 275, whereupon the planar crank member 283 (FIG. 6) and

the second rocker arm 47 (FIG. 1) are rotated in a counterclockwise direction. This action translates the valving element 52 of the flow control valve 53 (FIG. 3) rightwardly from its null position, and fluid under pressure is thereby caused to flow to the actuator 11 causing it to rotate the driving arm 240 and the load 241 in a clockwise direction. More specifically, fluid under pressure is caused to flow from the first pressure source 29 successively through the valve second outlet 93, the conduit portion 95A, the second restrictive orifice 235 of the first valving member 171, and the conduit portion 95B, to the chamber portion of the actuator 11 immediately to the right of the first piston 12 (and, correspondingly, from the second fluid source 30 to the second actuator piston 13), thus operating the actuator 11 to translate the driving arm 240 and the load 241 in a clockwise rotational direction. When the load 241 then reaches its preselected position, the valving element 52 of the flow control valve 53 is positioned in its null position by the linkage rod 253, the walking beam structure 262, idler crank 280, and the planar crank member 283, as has been described, whereupon the actuator 11 constrains the load 241 in the preselected position.

Detailed Description Text (93):
It will be recognized by those in the art that apart from the action of the fail-safe valving system 170 in restricting fluid flow between the actuator 11 and the flow control valve 53 upon the occurrence of a failure of the fluid amplifiers, any feedback system (not shown) which would operate the actuator 11 to move the load 241 could tend to move the load 241 in a violent and possibly hazardous manner, since the flow control valve 53 would be maintained in a "wide open" operative condition until the preselected position was reached. (In normal operation, the closedloop feedback system of the servo control system 26 provided by the position sensor system 16, the differential signal amplifiers 19, 19A, 19B, 19C, and the fluid amplifiers 21, 21A, 21B, and 21C prevent such violent operation of the actuator 11).

Detailed Description Text (94):
Thus, the fail-safe valving system 170 permits safe operation of the position feedback linkage system 58 to operate the flow control valve 53 and the actuator 11 to appropriately position the load apart from the operation of any of the electrical feedback elements of the servo control system 26, providing fail-safe, emergency operation of the actuator 11 even during the occurrence of a complete failure of all the electrical portions of the control system 26.

Detailed Description Text (95):
In the event of a failure of the fluid supply system upstream of the flow control valve 53 whereby the flow control valve 53 can no longer operate the actuator 11, the fail-safe valving system 170 is operative to shut off further fluid flow between the flow control valve 53 and the actuator 11, as has been previously discussed, thus locking the load 241 in its current position.

CLAIMS:

1. A fluid powered servo control system responsive to a plurality of redundant, command input signals and comprising:

a source of fluid under pressure;

a plurality of conduits communicating with the source of fluid under pressure;

a plurality of fluid amplifiers each operative in response to a respective command signal, each having a fluid inlet connected to a respective one of the conduits and first and second fluid outlets, and each comprising means receiving fluid from the source of fluid under pressure and ejecting fluid proportionally through the two outlets in response to a respective command signal to cause a differential

pressure output signal across the outlets, the fluid outlets of each fluid amplifier being fluidly isolated from the outlets of the other fluid amplifiers;

a housing;

a movable summing structure slideably mounted within the housing and comprising means summing the output signals of the fluid amplifiers, the movable summing structure having a plurality of pairs of piston face areas, each pair being fluidly associated with a respective fluid amplifier, each piston face area comprising means for receiving fluid pressure from a respective one of the fluid amplifier outlets;

a plurality of differential pressure monitoring means, each monitoring means being connected to a respective one of the conduits in series between the source of fluid under pressure and the corresponding fluid amplifier and comprising a means for shutting off fluid to the respective fluid amplifier upon the occurrence of a differential pressure, between the outlets of the respective fluid amplifier, which exceeds a predetermined level;

a flow control valve having a movable valve element, the summing structure being operatively connected to the movable valve element, the flow control valve being fluidly connected to, and receiving fluid under pressure from, the source of fluid under pressure;

a fluid powered actuator drivingly connected to a movable load, the flow control valve being fluidly operatively connected to the actuator;

a fail-safe valving system fluidly connected in series between the flow control valve and the actuator and comprising means for restricting any fluid flow between the flow control valve and the actuator upon concurrent operation of all the monitoring means shutting off fluid flow to all the fluid amplifiers, and for shutting off fluid flow between the flow control valve and the actuator upon the occurrence of a failure of the source of fluid under pressure whereby fluid under pressure is not supplied to the flow control valve.

2. The apparatus of claim 1, further comprising a position feedback system connected between the movable load and the movable valve element of the flow control valve and comprising means for actuating the flow control valve to operate the actuator to move the load structure to a preselected position within its positional range upon the fail-safe valving system being operative to restrict any fluid flow between the flow control valve and the actuator.

7. The apparatus of claim 3, wherein the control system third portion is a fluid powered actuator, the actuator being drivingly connected to a load structure movable within a range of movement, the control system second portion comprising a flow control valve fluidly operatively connected to the actuator, the apparatus including means translating the load to a preselected position within its positional range upon the valving structure being in its second position.

8. The apparatus of claim 7, the means translating the load to a preselected position comprising position feedback means, connected between the movable load structure and the flow control valve, for actuating the flow control valve to operate the actuator to move the movable load structure to the preselected position, within its positional range, upon the valving structure being moved to its second position, the position feedback means comprising means providing an error signal to the flow control valve until the movable structure is moved to the preselected position.

16. In combination with a servo control system of the type operable for positioning a movable structure

within a range of movement in response to a position command signal, having a flow control valve fluidly operatively connected to an actuator which is drivingly connected to the movable structure, the flow control valve being fluidly connected in series between a source of fluid under pressure and the actuator, the servo control system having a plurality of fluid amplifiers responsive to the position command signal to produce respective fluid output signals, means normally supplying fluid flow to each fluid amplifier and, alternatively, shutting off fluid flow to any fluid amplifier whose output signal falls outside a predetermined range, and means summing the fluid amplifier output signals and operatively transmitting a resultant signal to the flow control valve, a fail-safe valving system fluidly connected in series between the flow control valve and the actuator and comprising:

a housing defining at least one chamber;

a valving structure slideably seated within the at least one chamber and movable through a range of positions which include first, second, and third sequential positions, the valving structure having valving means preventing fluid flow between the flow control valve and the actuator upon the valving member being in its third position, having porting means permitting substantially unrestricted fluid flow between the flow control valve and the actuator upon the valving member being in its first position, and having orificial means permitting restricted fluid flow between the flow control valve and the actuator upon the valving member being in its second position;

means urging the valving member toward its third position;

means, having fluid communication with each fluid amplifier, for urging the valving member toward, and constraining the valving member in, its first position upon any of the fluid amplifiers receiving fluid under pressure from the means normally supplying fluid flow to each fluid amplifier, and for releasing the valving member from constraint upon the fluid supply to all the fluid amplifiers being shut off, whereby the valving member is moved from its first position by the means urging the valving member toward its third position; and

means normally preventing movement of the valving member from its second to its third position and, alternatively, permitting movement of the valving member to its third position upon the occurrence of a loss of fluid pressure upstream of the flow control valve.

18. The apparatus of claim 17, the fluid powered system third portion comprising an actuator, the actuator being drivingly connected to a movable load structure, the fluid powered system second portion comprising means fluidly operatively connected to the actuator, the apparatus further comprising:

a position feedback system connected between the movable load structure and the means fluidly operatively connected to the actuator and comprising means for actuating the means fluidly operatively connected to the actuator to operate the actuator to move the load structure to a preselected position upon the fail-safe valving system being operative to restrict any fluid flow between the second and third portions of the fluid powered system.

20. The apparatus of claim 18, the means fluidly operatively connected to the actuator comprising a flow control valve having a movable valve element, the position feedback system being operatively connected to the movable valve element.

21. The apparatus of claim 19, the means fluidly operatively connected to the actuator comprising a flow control valve having a movable valve element, the position feedback system comprising means operatively connected to the movable valve element.

22. The apparatus of claim 21; the position feedback system comprising a linkage system connected between the movable load structure and the movable valve element of the flow control valve, the linkage system including actuating means preventing operation of the linkage system to move the valve element of the flow control valve when the fail-safe safe valving system is operative to permit substantially unrestricted fluid flow between the second and third portions of the fluid powered system, and alternatively, operatively engaging the linkage system with the valve element of the flow control valve upon the fail-safe valving system being operative to restrict fluid flow between the second and third portions of the fluid powered system.

23. The apparatus of claim 22, the actuating means including an idler crank rotatably connected, upon a first pivot axis, to a crank member operatively connected to the movable valve element of the flow control valve and having a cam follower mounted on the idler crank at a location spaced from the first pivot axis, a cam member being provided rotatably mounted on the housing, further comprising means operative for bringing the cam member into engagement with the cam follower upon the fail-safe valving system being operative to restrict fluid flow between the second and third portions of the fluid powered system, the cam member and cam follower, when in their mutually engaged mode, comprising means preventing rotation of the idler crank about its first pivot axis and permitting rotation of the idler crank about a second pivot axis through the cam follower whereby the rotation of the idler crank is operable to translate the crank member and the valve element of the flow control valve.

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L1: Entry 66 of 413 File: USPT Jul 8, 2003

DOCUMENT-IDENTIFIER: US 6591201 B1
TITLE: Fluid energy pulse test system

Brief Summary Text (12):
Previous investigators have approached the problems of testing fluid control devices dynamically in a number of ways, several of which are briefly discussed. For example, U.S. Pat. No. 5,616,824 to Abdel-Malek et. al. (1997), teaches a valve diagnostic system for installed electromechanical control valves. The system identifies and compares to file data, a time-signature of valve operation to detect or predict potential valve failure. U.S. Pat. No. 5,524,484 to Sullivan (1996), teaches a diagnostic system for solenoid valves that are installed in line and which are in service. U.S. Pat. No. 5,329,956 to Marriott (1994), teaches a method of time signature analysis for electrically actuated, pneumatically controlled valves. U.S. Pat. No. 5,197,328 to Fitzgerald (1993), teaches a diagnostic method for pneumatically operated control valves. U.S. Pat. No. 5,272,647 to Hayes (1993), teaches a portable device that perturbs a valve actuator and monitors valve stem displacement, actuator pressure, and other valve parameters for steady state flow conditions. U.S. Pat. No. 4,903,529 to Hodge (1990), teaches a method and apparatus to analyze a hydraulic control valve and actuator assembly during plant shut-down periods. U.S. Pat. No. 4,893,494 to Hart (1990), teaches a method to evaluate safety valves removed from an installation by using either pneumatic or hydraulic fluids. U.S. Pat. No. 4,464,931 to Hendrick (1984), addresses steady state dynamic valve testing with an apparatus that checks a valve calibration (closing) pressure, checks valve closing integrity, and checks valve flow rate at a pressure greater than the valve calibration pressure.

Brief Summary Text (13):
The concept of a pressure pulse is derived from unsteady fluid flow and the propagation of large amplitude nonlinear waves, from which shock waves can be generated as a result of the physical attributes of non-steady-state fluid flow. An initially continuous waveform that advances into a uniform, stationary fluid is termed a pressure pulse. Pressure pulses are referenced in U.S. Pat. No. 4,549,715 to Engel (1985), which teaches an apparatus to generate gaseous pressure pulses to rapidly open an exhaust path to create a high volume, low pressure pulse. U.S. Pat. No. 4,686,658 to Davison (1987), teaches an apparatus and method for actuating a valve for imparting pressure pulses in a pressure pulse telemetry system wherein the actuating force is adjusted in response to a measured value of the minimum force necessary to actuate the valve. This provides a self-adjusting actuator to reduce the power needed to actuate the valve, prolong battery life in the associated batteries, and prolong valve and circuit life. And, U.S. Pat. No. 5,176,164 to Boyle (1993), teaches a flow control valve system for a hydrocarbon-producing well using gas-lift technology, in which the orifice size of a down-hole valve is electrically or pressure pulse controlled from the surface. The fluid flow rate through the valve is controlled over a continuous range, keeping the orifice size constant when necessary. Valve orifice size and well conditions are monitored down-hole and transmitted to the surface as feedback signals for valve control.

Detailed Description Text (8):
FIGS. 2A and 2B represent either electro-pneumatic fill valve assembly #132 or fill valve assembly #242, shown in FIG. 1. Regulated low pressure process fluid 168 is used to activate low pressure pneumatic actuator 180 which is connected by mechanical linkage 203 to high pressure ball valve 182. Fluid 168 opens and closes valve 182 to permit high pressure fluid flow from main storage tanks 20 into upstream or downstream reservoir storage tanks 30 or 46, through inlet pipe 208 and through outlet pipe 206. In FIG. 2A, fluid 168 is connected to voltage-to-pressure converter V/P#1184 (also, voltage-to-pressure switch) through low pressure pipe 186. When converter 184 is activated, low pressure fluid enters low pressure pneumatic actuator 180 through pipe 202. Actuator 180 is coupled to valve 182 by linkage 203.

Detailed Description Text (13):
In FIG. 1, upstream pressure transducer PT160, downstream pressure transducer PT268, upstream reservoir pressure transducer PT362, downstream reservoir pressure transducer PT472, and differential fluid pressure transducer DP 64 are in fluid communication with process fluid. These transducers are connected into the FEPTS apparatus by pipes and threaded pressure taps. These transducers are also connected by wiring to a signal conditioner unit 50 by signal transducer leads 104, 98, 92, 94, and 102, respectively. Differential fluid pressure transducer 64 is in fluid communication with process fluid and is connected by pipes 134 and 136 across a fluid control device within chamber 56, or across a fluid system substituted for chamber 56. Transducers 60, 68, 62, 72, and 64 are connected to electrical power supplies 86, 78, 82, 74, and 88 by leads 86A, 78A, 82A, 74A, and 88A.

Detailed Description Text (15):
Fluid flow rate transducer FT 70 is connected in line with pipes 140 and 142 to measure fluid flow exhausted to atmosphere 164. Alternative embodiments (not shown) connect transducer 70 in line with pipes 156 and 146 to measure fluid flow into downstream reservoir tanks 46 or in line with pipes 124 and 130 to measure fluid flow into chamber or system 56. Clearly, a plurality of fluid flow rate transducers could be installed in the FEPTS apparatus at these upstream and downstream locations. In practice, only one transducer 70 is required for fluid flow measurement when the apparatus is configured for single-input-single-output fluid flow. If an embodiment of the apparatus requires multiple input or multiple output fluid flows, additional fluid flow transducers are placed at appropriate inlet and outlet pipes. In the preferred embodiment, transducer 70 is connected to signal conditioning unit 50 by electrical lead 100. Transducer 70 is also connected to power supply 76 by lead 76A.

Detailed Description Text (19):
Valve control computer 170 is connected to control valve actuators in assemblies 32, 34, 36, 38, 40, 42, 44, and 48, as shown in FIG. 2A, FIG. 3A, and FIG. 4A.

Detailed Description Text (22):
In FIG. 3A, two methods of precise control of high energy fluid pressure pulses are illustrated. One method is by computer motor-positioning of a set ball valve 234. A second method is by manual motor-positioning of valve 234. Activating an assembly 266 requires setting valve 234 to a specific open setting and using ball valve 230 as a bang-bang actuator to allow passage of fluid through the assembly. Pipe 232 connects valve 230 to valve 234. Set valve position indicator #1237 provides an electrical signal through lead 229 to indicate valve position to a computer 170 and to a conventional digital display (not shown). Indicator 237 is linked by lead 235 to valve 234, so that a change in valve stem position is translated from a mechanical position into an electrical signal. This valve position signal is displayed on a computer monitor and a conventional digital display (not shown).

Detailed Description Text (24):

In FIG. 3A, computer control of valve 230 is achieved by connecting voltage-to-pressure converter V/P#2216 to computer 170 via computer protection diode #2218. Low pressure process fluid 168 is connected by pipe 214 to converter 216. When converter 216 is activated, low pressure fluid is sent through pipe 222 to pneumatic actuator 224, that is connected by mechanical linkage 226 to valve 230. When converter 216 is activated, valve 230 changes by a bang from fully closed to fully open. When converter 216 is deactivated, valve 230 changes by a bang from fully open to fully closed. Pneumatic actuator 224 is a conventional double-acting fluid cylinder.

Detailed Description Text (26):
In FIG. 3A, manual control of valve 230 is achieved by manual override switch 246 connected to power supply E 248 by lead 250 and thereafter to converter 216 by lead 247. Closing switch 246 activates pneumatic actuator 224, causing valve 230 to open fully. Opening switch 246 deactivates pneumatic actuator 224, causing valve 230 to close fully. Opening and closing valve 230 generates bang-bang fluid control.

Detailed Description Text (29):
In FIG. 4A, two methods of precise bang-bang control of high energy fluid pressure pulses are illustrated. In one method, electro-magnetic set valve 306 is motor-positioned by computer. In method two, valve 306 is motor-positioned manually. Activating assembly 314 requires setting valve 306 to a specific open setting and using solenoid pulse valve 308 as a bang-bang actuator to allow passage of fluid through the assembly. Valve 308 is connected to valve 306 by pipe 304. Valve position indicator #2311 provides an electrical signal through lead 307 indicating valve position to computer 170 and to a conventional digital display (not shown). Indicator 311 is connected to valve 306 by lead 309 so that a change in valve stem position is translated from a mechanical position into an electrical signal. This valve position signal is displayed on a computer monitor and a conventional digital display (not shown).

Detailed Description Text (45):
FIG. 6E shows a bottom insert 460 that is screwed into lower chamber body 440 of test chamber 56 (FIG. 6A) to seat GLVs within the test chamber. Upper external threads 461 of bottom insert 460 attach at lower internal threads 442 of lower chamber body 440. Bottom insert 460 comprises bottom insert "O" ring 470, "O" ring recess 471, first and second GLV "O" rings 472 and 474, first and second GLV "O" ring recesses 473 and 475, third and fourth pressure tap holes 465 and 469, seating ridge 476, upper fluid path 477, center fluid path 478, lower fluid port 479, and lower port internal threads 480. When bottom insert 460 is screwed into lower chamber body 440, external pressure integrity is achieved with body 440 by "O" ring 470. Internal pressure integrity for fluid passing through a GLV within test chamber 56 is achieved by "O" rings 472 and 474. Pressure taps 462 and 466 permit pressure transducer and differential pressure transducer connections to be made downstream of a GLV within test chamber 56.

Detailed Description Text (48):
FIG. 7A shows a 2.54 centimeter (one inch) diameter conventional tubing retrievable gas-lift valve [TR-GLV] 500 with a 2.54 centimeter (one inch) conventional check valve housing 508 screwed onto the base of the TR-GLV at threads 503. Housing 508 is shown with lower threads 509 and outlet port 510. Top threaded bolt 501 can be removed in order to set TR-GLV 500 opening pressure. When TR-GLV 500 is installed in test chamber 56, top screw cap 411 is screwed onto upper test chamber body 400, and hammer union nut 416 is tightened. Further tightening of cap 411 causes forcing member 405 to generate an axial force on bolt 501 to hold the TR-GLV in place in test chamber 56 (see FIG. 6A). A high-pressure seal between housing 508 and bottom insert 460 is created by contact of housing 508 with "O" rings 472 and 474, when housing 508 rests on seating ridge 476 (see FIG. 6A). Fluid enters TR-GLV 500 at valve inlet port 502 and leaves at housing outlet port 510. Valve inlet port 502 is

upstream, and housing outlet port 510 is downstream, of a conventional TR-GLV 500 valve stem and valve seat (not shown).

Detailed Description Text (49):
FIG. 7B shows a 3.81 centimeter (one and one-half inch) diameter conventional tubing retrievable gas-lift valve [TR-GLV] 504 with a 2.54 centimeter (one inch) conventional check valve housing 508 screwed onto the base of the TR-GLV at threads 507. Housing 508 is shown with lower threads 509 and outlet port 510. Top threaded bolt 505 can be removed in order to set TR-GLV 504 opening pressure. When TR-GLV 504 is installed in test chamber 56, top screw cap 411 is screwed onto upper test chamber body 400, and hammer union nut 416 is tightened. Further tightening of cap 411 causes forcing member 405 to generate an axial force on bolt 505 to hold the TR-GLV in place in test chamber 56 (see FIG. 6A). A high-pressure seal between housing 508 and bottom insert 460 is created by contact of housing 508 with "O" rings 472 and 474, when housing 508 rests on seating ridge 476 (see FIG. 6A). Fluid enters TR-GLV 504 at valve inlet port 506 and leaves at housing outlet port 510. Valve inlet port 506 is upstream, and housing outlet port 510 is downstream, of a conventional TR-GLV 504 valve stem and valve seat (not shown).

Detailed Description Text (55):
FIG. 8B shows a 3.81 centimeter (one and one-half inch) diameter conventional WR-GLV 550, with external components, including threaded bolt 551, latch threads 552, inlet port 554, and lower threads 555; and with internal components, including nitrogen chamber 564, nitrogen chamber fill access port 562, bellows 566, valve stem 568, internal valve 570, valve seat 571, and valve seat orifice 572. FIG. 8B also shows WR-GLV 550 with a 3.81 centimeter (one and one-half inch) diameter conventional check valve assembly 573 attached at WR-GLV threads 555. Check valve assembly 573 includes check valve 575, valve seat 574, and valve spring 577; housing 576 with lower threads 580, and cap 579, with outlet port 578, attached at threads 580.

Detailed Description Text (114):
FIG. 11A shows data for a test of a typical injection pressure operated GLV [IPO-GLV] used in petroleum production. The IPO-GLV opens by upstream pressure on the IPO-GLV bellows when this pressure overcomes the threshold force generated by the IPO-GLV's pressurized bellows which keeps the IPO-GLV internal valve closed on its valve seat (see FIG. 8B). The IPO-GLV begins to open when movement of the bellows pulls the IPO-GLV internal valve away from the IPO-GLV valve seat. The IPO-GLV closes when upstream pressure on the bellows is reduced below the opening pressure threshold.

Detailed Description Text (118):
This additional opening pressure is needed to overcome internal static friction forces that include bellows tension and surface tension between the internal valve and valve seat in the IPO-GLV. This feature is shown in upstream pressure path 602 and differential pressure path 604. In path 602, the upstream pressure reaches a maximum 7,134 kPa (1020 psig) at time 0.45 seconds and then decays on path 602 to a steady state opening pressure of 6,306 kPa (900 psig) before closing at time 1.45 seconds. In path 604, the differential pressure between upstream pressure and downstream IPO-GLV pressure also reaches a maximum at time 0.45 seconds. In path 604, the increasing differential pressure from its minimum value is a function of the downstream open-to-the-atmosphere piping configuration from the IPO-GLV test chamber. [2] IPO-GLVs do not close instantaneously, as shown by the decay of upstream pressure path 602, starting at time 1.45 seconds when the pulse is terminated, to a closing pressure of 5,961 kPa (850 psig) at 1.5 seconds. This closing feature is also shown by the relative minimum of path 604 at time 1.5 seconds. When the pulse is terminated at 1.45 seconds, downstream IPO-GLV exhaust pressure decreases more rapidly than the decrease in upstream IPO-GLV pressure. [3] Upstream

reservoir pressure path 600 shows the 0.4 cubic foot volume upstream reservoir tank initial pressure condition of 9,616 kPa (1380 psig). When the input pulse is initiated at time 0.35 seconds, the rate of upstream reservoir pressure decay is determined by the time constants associated with the FEPTS apparatus and the IPO-GLV. These time constants are directly related to the upstream reservoir apertures, pipes, and volume; input and exhaust set valves apertures; test chamber upstream and downstream pipes, and, the IPO-GLV in the test chamber. [4] The upstream reservoir time constant is the slowest (that is, longest) time constant associated with the FEPTS apparatus. It is determined by a large reservoir volume and a relatively small aperture of the input function generating set valve. This slow reservoir time constant is desirable in order to generate a nearly linear decrease in upstream reservoir pressure during a test. [5] In engineering practice, the measurement accuracy of time dependent variables is enhanced when the time constant of a driving function is faster (that is, shorter) by a factor or ten or more than the slowest (that is, longest or dominant) time constant of the system under investigation. The dominant time constant of a typical IPO-GLV is a function of the mechanical and pneumatic properties of the bellows, including bellows volume, bellows pressure charge, and bellows surface area for incoming fluid contact. This dominant IPO-GLV time constant is illustrated in FIG. 11A by the decay from a peak opening pressure at time 0.45 seconds on path 602 to a steady state opening pressure at approximate time 1.0 seconds, which represents five time constants of duration 0.55 seconds. Therefore, the IPO-GLV opening pressure time constant is 0.110 seconds. In the FEPTS apparatus, tests have shown that the shortest rise time constant of a generated pulse can be controlled in a range of 0.005 seconds to 0.010 seconds, which satisfies the standard engineering measurement criterion. The shortest rise time constant of an energy pulse applied by the apparatus is between 11 and 22 times faster than the dominant time constant of a typical IPO-GLV. [6] The flow rate through the IPO-GLV is shown in FIG. 11 A by path 606. The maximum flow rate is 450 MSCF/D when the IPO-GLV is driven quickly to an open state with upstream pressure higher than the pressure required to keep the IPO-GLV open. Flow rate through the valve decreases at a constant rate from approximately 1.0 seconds to IPO-GLV closure at 1.45 seconds. Just prior to IPO-GLV closure at 5,961 kPa (850 psig), the fluid flow rate is 275 MSCF/D. Lower flow rates would be shown by a longer duration energy pulse, for example, a pulse of two second duration. The linear portion of fluid flow rate path 606 can be extrapolated to zero flow rate. Zero flow rate occurs when upstream reservoir pressure reaches IPO-GLV closing pressure of 5,961 kPa (850 psig) that occurs at a time of about 3.0 seconds. [7] The upstream fluid pressure path 602 and fluid flow rate path 606 permit accurate determination of upstream reservoir pressure needs to achieve a particular flow rate through the IPO-GLV. The flow rate through, the opening pressure of, and the closing pressure of the IPO-GLV are important variables that are used in the design of a GLV string for unloading wells and for lifting petroleum products from underground formations. The FEPTS data shown in FIG. 11A demonstrate that GLV test data can be used for gas-lift lifting designs. [8] A method for designing IPO-GLV operating conditions by using FEPTS test data is illustrated in FIG. 11A by choosing a desired flow rate on path 606, drawing a pressure intersecting design line 603 vertically through the graph to intersect upstream reservoir pressure path 600 and upstream IPO-GLV pressure path 602. The pressure corresponding to the intersection point on path 600 provides the upstream reservoir pressure (generated by a field compressor) that is required to operate the IPO-GLV at an open pressure corresponding to the intersection point on path 602 for the chosen flow rate. For example, a flow rate of 300 MSCF/D on flow rate path 606 is selected and is shown by time flow rate design line 601, which intersects path 606. Design line 603 is drawn vertically to intersect path 600 and path 602. The upstream reservoir pressure required to deliver 300 MSCF/D through the IPO-GLV is 7,686 kPa (1100 psig), as shown by upstream reservoir pressure design line 605. For this example, an upstream IPO-GLV operating pressure of 6,303 kPa (900 psig) is shown by the intersection of line 603 and path 602. [9] It is generally known that fluid flow through a valve assembly, when the internal valve is fully open, is different from fluid flow through the same valve assembly when the internal valve begins to move away from (that is, to open) or move toward (that is, to close) its internal valve seat. The opening and closing properties of fluid flow through a valve and the pressure generating this fluid flow can be

evaluated from time-dependent paths 600, 602, 604, and 606, by expanding the time axis to include greater detail of the pressure-flow data.

Detailed Description Text (121):
The parametric presentation of data in FIG. 11B also shows transient flow rates that occur while the IPO-GLV internal valve is moving away from its valve seat (that is, is opening) and moving toward its valve seat (that is, is closing). The curvature of upstream pressure path 610 with respect to flow rate before reaching point 614 shows the IPO-GLV opening rapidly with a corresponding high rate of increase of flow rate. In a similar way, the curvature of upstream pressure path 610 with respect to flow rate after leaving point 614 shows the IPO-GLV closing with a corresponding high rate of decrease of flow rate. Parallel tangents (not shown) to points on path 610 during the initial stage of opening and the initial stage of closing can show that increasing and decreasing rates of change of pressure with respect to a change of flow rate are equal for different flow rates. This result can be interpreted as a smoothly operating internal bellows during IPO-GLV opening and closing transients. Stated differently, there are no apparent internal binding constraints and no apparent excessive internal friction acting on the bellows.

Detailed Description Text (142):
FIGS. 14A, B, C, and D, show dynamic performance characteristics of a typical production pressure operated GLV [PPO-GLV] installed in test chamber 56. PPO-GLVs are also called fluid valves, a term that defines a valve geometry in which an internal reverse flow check valve prevents fluid from moving upstream through the PPO-GLV. PPO-GLVs are designed with an internal valve spring, a bellows, or combined spring-bellows to hold the PPO-GLV internal valve on its valve seat. The internal reverse flow check valve and the need to open a PPO-GLV with downstream pressure require controlling the rate of increase of downstream pressure to open, and to keep open, the PPO-GLV. When the downstream pressure is greater than a threshold pressure that keeps the PPO-GLV internal valve closed on its valve seat, the PPO-GLV opens. When the downstream rate of increase of fluid flow is greater than a threshold downstream rate of increase of fluid flow, which is observable and controllable by the downstream rate of pressure increase, the PPO-GLV internal valve opens and quickly closes because the internal reverse flow check valve is activated and closes.

Detailed Description Text (145):
Prior to the test, downstream pressure is shunted past the PPO-GLV internal valve orifice in order to generate pressure on the internal spring and bellows and, thus, to put the PPO-GLV in a fully open condition. The PPO-GLV must be fully open at the start of the test so that the flow rate through this valve can be determined. The PPO-GLV remains open as long as the force generated by downstream pressure is greater than the spring-bellows threshold force that keeps the PPO-GLV internal valve closed on its valve seat. The PPO-GLV closes when downstream pressure is reduced below the opening pressure threshold. The downstream pressure is a function of the upstream-to-downstream fluid flow through the PPO-GLV.

CLAIMS:

4. The apparatus of claim 3 wherein a pressure transducer is in fluid communication with main fluid storage reservoir, a pressure transducer is in fluid communication with upstream fluid storage reservoir, a pressure transducer is in fluid communication with downstream fluid storage reservoir, a pressure transducer is in fluid communication with upstream test chamber or fluid system, a pressure transducer is in fluid communication with downstream test chamber or fluid system, a differential pressure transducer is in fluid communication and connected across said test chamber or fluid system, a temperature transducer is in fluid communication upstream and a temperature transducer is in fluid

communication downstream of said test chamber or fluid system, and a fluid flow rate transducer is in fluid communication upstream or downstream of said test chamber or fluid system.

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L1: Entry 148 of 413 File: USPT Nov 16, 1999

DOCUMENT-IDENTIFIER: US 5984349 A
TITLE: Low profile hydraulic seat weight sensor

Abstract Text (1):
A hydrostatic weight sensor comprises one or more fluid containing bladders in fluid communication with one another and with a pressure sensor, each constructed from a plurality of sheets of semi-rigid material sealably connected at the periphery thereof so as to form one or more enclosed volumes, and filled with a fluid so as to cause deformation of the bladder faces. The hydrostatic weight sensor is located within the seat, and the pressure of the fluid contained therein is responsive to the applied load. In another embodiment, a plurality of sheets of semi-rigid material are sealed at a periphery and are secured to one another at a plurality of locations within the periphery so as to create a plurality of fluid containing zones which can hinge with respect to one another. The sensing fluid is either a liquid, a Bingham plastic, or a thixotropic material.

Parent Case Text (3):
Co-pending U.S. application Ser. No. 08/933,701, hereinafter "Application ASL-157-US", entitled "Seat Weight Sensor Having Fluid Filled Bladder", filed on Dec. 18, 1997 claiming benefit of U.S. Provisional Application Ser. No. 60/032,380 filed on Dec. 19, 1996, and assigned to the assignee of the instant invention discloses a hydrostatic weight sensor comprising a fluid filled bladder and a pressure sensor for sensing the weight of an occupant in a vehicle seat for controlling a safety restraint system. Application ASL-157-US also discloses a load distributor for distributing loads across the load bearing surface of the hydrostatic weight sensor.

Parent Case Text (4):
Co-pending U.S. application Ser. No. 09/033,672, hereinafter "Application ASL-161-US", entitled "Automotive Seat Weight Sensing System", filed on Jan. 7, 1998 claiming benefit of U.S. Provisional Application Ser. No. 60/034,018 filed on Jan. 8, 1998, and assigned to the assignee of the instant invention discloses a seat weight sensing system comprising a plurality of hydrostatic weight sensors each of which is in accordance with Application ASL-157-US.

Parent Case Text (5):
Co-pending U.S. application Ser. No. 09/003,870, hereinafter "Application ASL-163-US", entitled "Vehicle Seat Sensor Having Self-Maintaining Air Bladder", filed on Jan. 7, 1997 claiming benefit of U.S. Provisional Application Ser. No. 60/035,343 filed on Jan. 16, 1997, and assigned to the assignee of the instant invention discloses an apparatus for automatically maintaining the supply of sensing fluid in a hydrostatic weight sensor.

Parent Case Text (6):
Co-pending U.S. application Ser. No. 09/033,851, hereinafter "Application ASL-185-US", entitled "Seat Weight Sensor Having Self-Regulating Fluid Filled Bladder", filed on Jan. 7, 1998 claiming benefit of

U.S. Provisional Application Ser. No. 60/058,086 filed on Sep. 5, 1997, and assigned to the assignee of the instant invention discloses a hydrostatic weight sensor having a means for automatically regulating the amount of sensing fluid therein.

Parent Case Text (7):
Co-pending U.S. application Ser. No. 09/003,868, hereinafter "Application ASL-186-US", entitled "Seat Weight Sensor with Means for Distributing Loads", filed on Jan. 7, 1998 claiming benefit of U.S. Provisional Application Ser. No. 60/058,084 filed on Sep. 5, 1997, and assigned to the assignee of the instant invention discloses a load distributor for distributing sensed load across the load bearing surface of a hydrostatic weight sensor.

Parent Case Text (8):
Co-pending U.S. application Ser. No. 09/003,673, hereinafter "Application ASL-187-US", entitled "Seat Weight Sensor Having Self-Regulating Fluid Filled Bladder", filed on Jan. 7, 1998 claiming benefit of U.S. Provisional Application Ser. No. 60/058,119 filed on Sep. 5, 1997, and assigned to the assignee of the instant invention discloses a hydrostatic weight sensor having a means for automatically regulating the amount of sensing fluid therein.

Parent Case Text (10):
Co-pending U.S. application Ser. No. 09/003,746, hereinafter "Application ASL-194-US", entitled "Seat Weight Sensor Using Fluid Filled Tubing", filed on Jan. 7, 1998 claiming benefit of U.S. Provisional Application Ser. No. 60/065,986 filed on Nov. 17, 1997, and assigned to the assignee of the instant invention discloses a hydrostatic weight sensor incorporating a fluid filled tube.

Brief Summary Text (2):
The instant invention generally relates to sensors and systems for measuring weight and more particularly to a weight sensor for measuring the weight of occupants and other objects in a motor vehicle seat such as useful for determining occupant seating conditions for controlling a vehicle safety restraint system.

Brief Summary Text (4):
A vehicle may contain automatic safety restraint actuators which are activated responsive to a vehicle crash for purposes of mitigating occupant injury. Examples of such restraint actuators include air bags, seat belt pretensioners, and deployable knee bolsters.

Brief Summary Text (5):
One objective of an automatic safety restraint system is to mitigate occupant injury, thereby not causing more injury with the automatic restraint system than would be caused by the crash had the automatic restraint system not been activated. Notwithstanding the protective benefit of these automatic safety restraint actuators, there is generally both a risk and a cost associated with the deployment thereof. Generally, it is desirable to only activate automatic safety restraint actuators when needed to mitigate injury because of the expense of replacing the associated components of the safety restraint system, and because of the potential for such activations to harm occupants. This is particularly true of air bag restraint systems, wherein occupants too close to the air bag at the time of deployment--i.e. out-of-position occupants--are vulnerable to injury or death from the deploying air bag even when the associated vehicle crash is relatively mild. Moreover, occupants who are of small stature or with weak constitution, such as children, small adults or people with frail bones are particularly vulnerable to injury induced by the air bag inflator. Furthermore, infants properly secured in a normally positioned rear facing infant seat (RFIS) in proximity to a front seat passenger-side air bag are also vulnerable to injury or death from the deploying air bag because of the close proximity of the infant seat's rear surface to the

air bag inflator module.

Brief Summary Text (6):
While air bags are designed to protect vehicle occupants, conventional crash detection and safety restraint deployment systems only use sensors which are mounted on the vehicle frame and are triggered by acceleration or velocity of the car rather than the occupant. Accordingly, conventional deployment strategies are not directly based on the weight, stature, and position of vehicle occupants. It is often very difficult to discriminate between crashes where air bags should be deployed and when their deployment could cause more harm than benefit. This difficult decision is typically made using only one or as few as possible sensors mounted on the vehicle. In the future, more occupant safety strategies will be available, including seat belt pre-tensioning and multi-stage air bags. With more available options, the deployment decision will become more complicated and require additional real-time occupant position data.

Brief Summary Text (12):
Except for some cases of oblique or side-impact crashes, it is generally desirable to not activate an automatic safety restraint actuator if an associated occupant is not present because of the otherwise unnecessary costs and inconveniences associated with the replacement of a deployed air bag inflation system. Occupant presence can be detected by a seat weight sensor adapted to provide either a continuous measure of occupant weight or to provide a binary indication if the occupant weight is either above or below a specified weight threshold.

Brief Summary Text (13):
Known seat weight sensors comprise one or more pads employing force sensitive resistive (FSR) films. These arrangements are typically used as weight threshold systems to disable a passenger air bag when the seat is empty. Load cells attached to the seat mounting posts have also been used in research applications. Mechanisms which use string based potentiometers to measure downward seat displacement have also been investigated.

Brief Summary Text (14):
Such known arrangements suffer from several drawbacks. First, variable resistance force sensors have limited sensitivity and in some situations are not sensitive enough to put directly under a seat pad while still achieving the desired response. Second, the threshold weight system provides only very limited information. For example, such arrangements provide no indication as to the size of an occupant. Third, the resistance values of known variable force resistor change with temperature, and are subject to drift over time with a constant load on the sensor.

Brief Summary Text (15):
Furthermore, other known sensing arrangements do not otherwise provide suitable results. For example, the use of load cells is prohibitively expensive for large-scale commercial applications. Strain gauges of any type may be impractical because of the difficulty in applying them to the strained material. Mechanical string potentiometer based weight sensors are complex, and subject to failure from stretching of the string. String potentiometer based weight sensors also suffer from a limitation whereby seat geometry changes over the lifetime of the seat. More specifically, seats tend to take a "set" over time so that the springs and cushion tend to move downward as the seat ages. A string potentiometer based weight sensor measuring downward displacement would require periodic recalibration over the lifetime of the seat. Finally, optical or infrared sensors have been used to measure the spatial position of occupants relative to the dashboard or headliner. Often these sensors are also integrated with speed sensors to discern changes in occupant position due to car acceleration. Current optical and infrared occupant position sensors require augmented information from speed and weight sensors, thereby resulting in a relatively high cost distributed system which may be difficult to manufacture, install, and

maintain. Furthermore, optical and/or infrared sensors which measure the range from the headliner or dashboard can be confused by placement of objects in front of an occupant, such as when reading newspapers or books, or by the position of the seat back because many seats can recline fully back and incline fully forward. Moreover, the sensing aperture of these sensors may become occluded by inadvertent scratching or substance application.

Brief Summary Text (16):
Known seat weight sensing techniques generally require multiple points for sensing distributed weight accurately. Also, force sensing resistors, load cells or membrane switches may require significant seat redesign for use in current or future seats. This is particularly true for spring type seats which do not provide a uniform horizontal support surface. The response time of load cells or membrane switches may fast enough for realtime applications.

Brief Summary Text (17):
The prior art also teaches the use of seat weight sensors outside the automotive environment, for example as a means for disabling the activation of either a boat or an industrial machine if the operator is not properly seated, or for weighing a person seated on an exercise bike. These devices employ pneumatic bladders located in the seat, whereby the pressure within the bladder is used to either activate a threshold switch or to provide a continuous indication of occupant weight.

Brief Summary Text (18):
One problem with prior art pneumatic sensors, particularly when applied to the automotive environment, is their sensitivity to environmental conditions, particularly to ambient temperature and pressure. This requires the bladder to be partially filled with fluid under ambient conditions of lower temperature or higher pressure, thereby making the bladder more susceptible to bottoming out when exposed to localized or concentrated loads and therefor requiring a means for distributing the loads over the load bearing area of the bladder. Pneumatic seat weight sensors can be sensitive to the amount of air initially in the associated bladder. A seat weight sensor in an automotive environment must function reliably and accurately over a wide range of temperatures and pressures which can cause significant errors.

Brief Summary Text (19):
Another problem with a pneumatic seat weight sensor is that the sensor bladder must be sufficiently thick to prevent the top and bottom surfaces of the bladder from compressing against one another responsive to a sufficiently great localized or concentrated load under conditions when the bladder has a relatively small amount of gas, such as would occur when the bladder is filled at low pressure or high temperature.

Brief Summary Text (20):
Yet another problem with a pneumatic seat weight sensor is that a gas filled bladder is also prone to loss of fluid by leakage or osmosis, which necessitates a means for replenishing the working fluid of the bladder over the life of operation.

Brief Summary Text (21):
The prior art also teaches the use of hydraulic load cells, wherein the weight to be measured acts upon a piston element of known area, whereby the measured weight is found by multiplying a measured pressure times the known area. One problem with hydraulic load cells in the automotive environment, particularly in a seat, is that the effects of load cell orientation on hydraulic head can introduce load measurement errors.

Brief Summary Text (23):

The instant invention overcomes the above noted problems by providing a low profile hydraulic hydrostatic weight sensor constructed from a pair plates or sheets of semi-rigid material which is peripherally sealed to form a sealed area. The sealed area is in fluid communication with a sensing port. A sensing fluid is injected into the sealed area causing the planar walls of the sealed area to bulge outwardly thereby forming a sealed cavity. The sensing fluid is preferably either a liquid, grease, Bingham fluid, or a thixotropic material, preferably with a relatively small thermal expansion coefficient and retaining fluid-like properties over the range of temperatures which can be encountered in an automotive environment. A differential pressure transducer in fluid communication with the sensing fluid via the sensing port senses the pressure differential between the sensing fluid and the surrounding environment.

Brief Summary Text (24):
The above described hydraulic hydrostatic weight sensor is embedded within the seat, preferably below the seat cushion and above the seat springs. In operation, a load applied to the seat is transferred to the top of the hydraulic hydrostatic weight sensor via the seat cushion and reacted by the seat springs against the bottom of the hydraulic hydrostatic weight sensor, thereby causing the top and bottom surfaces of the hydraulic hydrostatic weight sensor to be compressed, compressing the sensing fluid of the hydraulic hydrostatic weight sensor. The pressure of the sensing fluid transfers the load from the top surface of the hydraulic hydrostatic weight sensor to the bottom surface thereof. Therefore, assuming that the arrangement of the hydraulic hydrostatic weight sensor within the seat is such that all of the load on the seat is supported by the hydraulic hydrostatic weight sensor, then the magnitude of the applied load is substantially given by the product of the sensor fluid pressure times the area of the bottom surface of the hydraulic hydrostatic weight sensor exposed to the sensing fluid.

Brief Summary Text (25):
One problem with a hydraulic hydrostatic weight sensor constructed with a semi-rigid material forming a single cavity is that the resulting assembly is relatively inflexible, especially in comparison with the flexibility of the associated seat spring. Such a hydraulic hydrostatic weight sensor when incorporated into a seat assembly could possibly reduce the seating comfort as perceived by a seated occupant because a relatively inflexible hydraulic hydrostatic weight sensor could decrease the overall flexibility of the seat. Alternately, the flexing of such a relatively inflexible hydraulic hydrostatic weight sensor by the motion of the seat cushion responsive to a seated occupant could result in a premature failure of the hydraulic hydrostatic weight sensor, for example due to fatigue and localized stress risers.

Brief Summary Text (26):
A second embodiment of the instant invention provides for a hydraulic hydrostatic weight sensor with increased flexibility of by incorporating into a single assembly a plurality of relatively small single cell embodiments which are in fluid communication with one another, whereby the interconnections between the separate cells are sufficiently flexible so as to enable the entire assembly to conform to the deflections of the seat cushion or seat spring. The multi-cell embodiment may be constructed as an assemblage of distinct single cells. Alternately the multi-cell embodiment may comprise a unitary construction incorporating a continuous sheet or plate for each of the top and bottom surfaces of the hydraulic hydrostatic weight sensor, whereby the assembly is partitioned into a plurality of cells with seams or tufts by which the cells adjacent thereto may hinge and which are arranged to provide for fluid communication amongst the cells.

Brief Summary Text (29):
The instant invention provides a seat weight sensor which can be constructed relatively inexpensively and ruggedly from steel stampings which are welded to form the associated cells. The cells are then shaped by filling with an appropriate sensing fluid under pressure. Thereafter, the cells retain their

deformed shape, which without the sensing fluid contained therein is relatively compliant under compressive loads applied to the faces of the cells. A unitary constructed multi-cell embodiment may alternately incorporate spring steel stampings thereby enabling the hydraulic hydrostatic weight sensor to also fimction as a seat spring.

Brief Summary Text (30):
Accordingly, one object of the instant invention is to provide an improved seat weight sensor which provides a consistent and accurate measure of the seat loading independent of the location of the source of weight on the seat.

Brief Summary Text (31):
A further object of the instant invention is to provide an improved seat weight sensor which provides a consistent and accurate measure of the seat loading independent of the size and distribution of the source of weight on the seat.

Brief Summary Text (32):
A yet further object of the instant invention is to provide an improved seat weight sensor which provides a consistent and accurate measure of the seat loading independent of the amount of weight on the seat.

Brief Summary Text (33):
A yet further object of the instant invention is to provide an improved seat weight sensor which operates over a wide range of ambient temperature and pressure conditions.

Brief Summary Text (34):
A yet further object of the instant invention is to provide an improved seat weight sensor which can distinguish between a rear facing infant seat, for which an air bag system is preferably not deployed, and other occupants for which an air bag system is preferably deployed in the event of a crash of sufficient severity.

Brief Summary Text (35):
A yet further object of the instant invention is to provide an improved seat weight sensor which can be incorporated into an intelligent safety restraint system for which the preferable mode of the activation of a controllable occupant restraint system is dependent upon the weight of the occupant.

Brief Summary Text (36):
A yet further object of the instant invention is to provide an improved seat weight sensor which does not interfere with occupant comfort.

Brief Summary Text (37):
A yet further object of the instant invention is to provide an improved seat weight sensor which is insensitive to the orientation of the seat.

Brief Summary Text (38):
A yet further object of the instant invention is to provide an improved seat weight sensor which is inexpensive to produce.

Brief Summary Text (39):
In accordance with these objectives, one feature of the instant invention is a low profile hydraulic hydrostatic weight sensor mounted in the seat.

Brief Summary Text (42):
Yet another feature of the instant invention is the incorporation of the hydraulic hydrostatic weight sensor below the seat cushion wherein the seat cushion acts to distribute the seat load across the surface of the hydraulic hydrostatic weight sensor.

Brief Summary Text (47):
The specific features of the instant invention provide a number of associated advantages. One advantage of the instant invention with respect to the prior art is that the hydraulic hydrostatic weight sensor is responsive to loads over a large area of the seat without regards to the distribution or amount of loading.

Brief Summary Text (48):
Another advantage of the instant invention is that the hydraulic hydrostatic weight sensor is relatively insensitive to variations in ambient pressure or temperature so that the seat weight sensor works consistently and accurately over a wide range of ambient pressures and temperatures.

Brief Summary Text (49):
Yet another advantage of the instant invention is that through the incorporation of a grease, Bingham fluid or thixotropic material as a sensing fluid, which in the hydraulic hydrostatic weight sensor does not exhibit a substantial hydrostatic head, the associated differential pressure measurement is relatively insensitive to the orientation of the seat.

Brief Summary Text (52):
Yet another advantage of the instant invention is that through the incorporation of spring steel in the surfaces of the hydraulic hydrostatic weight sensor, the instant invention can also function as a seat spring.

Brief Summary Text (53):
Yet another advantage of the instant invention is that the seat weight sensor thereof can enable a rear facing infant seat to be distinguished from an occupant for which the air bag system is preferably deployed.

Brief Summary Text (54):
Yet another advantage of the instant invention is that the seat weight sensor thereof is sufficiently robust and accurate to enable associated occupant weight dependent control of a controllable occupant restraint system.

Brief Summary Text (55):
Accordingly, the instant invention provides an improved seat weight sensor which is relatively insensitive to the effects of ambient temperature and pressure; which is simple in construction and relatively robust and reliable in operation; which has a low physical profile and which can be readily incorporated into an automotive seat without interfering with occupant comfort; and which can be produced relatively inexpensively.

Detailed Description Text (2):
Referring to FIG. 1, a seat 3 in a motor vehicle 1 incorporates a hydraulic hydrostatic weight sensor 10 mounted in the seat base 40. The hydraulic hydrostatic weight sensor 10 comprises hydraulic load cell element 15, or bladder, and a differential pressure sensor 20 for measuring the difference in pressure between the hydraulic load cell element and the atmosphere 25. The hydraulic load cell element 15 is sandwiched between the seat frame 46 below and the seat cushion foam 44 above.

Detailed Description Text (3):
In operation, an occupant 5 seated on the base 40 of seat 3 causes the pressure inside the hydraulic load cell element 15 to increase such that that product of the differential pressure, as sensed by differential pressure sensor 20, multiplied times the area of the base 17 of the hydraulic load cell element 15 is substantially equal to the total weight distributed by the seat cushion foam 44 over the top 19 of the hydraulic load cell element 15. The pressure signal output 22 from differential pressure sensor 20 is operatively coupled to a signal processor 50 which converts the pressure signal output 22 to a measure of occupant weight using known analog, digital, or microprocessor circuitry and software. A crash sensor 60 is also operatively coupled to the signal processor 50. Responsive to a crash detected by the crash sensor 60, and further responsive to the sensed weight of the occupant as transformed from the pressure signal output 22, the signal processor 50 generates a signal 80 which is operatively coupled to one or more initiators 90 of one or more gas generators 100 mounted in an air bag inflator module 110, thereby controlling the activation of the air bag inflator module assembly 7 so as to inflate the air bag 120 as necessary to protect the occupant 5 from injury which might otherwise be caused by the crash. The electrical power necessary to carry out these operations is provided by a source of power 70, preferably the vehicle battery.

Detailed Description Text (9):
By the incorporation of either a grease, Bingham fluid or thixotropic material as the sensing fluid within the cavity of the hydraulic load cell element 15 the orientation thereof has a relatively small effect on the differential pressure signal. More particularly, a Bingham fluid, also known as a Bingham plastic, acts as a solid when the shear stress therein are below a threshold, and acts a Newtonian fluid otherwise. Furthermore, a material such as grease attaches to the inside surfaces of the hydraulic load cell element 15 and does not flow in response to a change in the orientation thereof. Therefore, the differential pressure signal from a hydraulic load cell element 15 incorporating such a sensing fluid and incorporated in a vehicle seat would be relatively insensitive to the orientation of the seat.

Detailed Description Text (10):
The hydraulic load cell element 15 incorporated in a hydraulic hydrostatic weight sensor 10 must have sufficient flexibility so as to conform to the profile of the seat responsive to loading by the occupant. A single load element constructed in accordance with FIGS. 1-5 and of sufficient area to serve as a seat weight sensor would be relatively stiff in bending because of the expanded shape of the single hydraulic load cell element 15.

Detailed Description Text (11):
Referring to FIG. 6, a hydraulic load cell element 15 with improved flexibility comprises a plurality of cells 602 in constructed in accordance with the FIGS. 1-5 and in fluid communication with one another via pressure distribution ports 18. A load applied across the top surfaces 19 of the individual cells 602 increases the pressure of the sensing fluid within the hydraulic load cell element 15 in accordance with equation (1), where DP is the pressure differential pressure between the sensing port 16 and the atmosphere 25, and A is the composite projected area of all cells 602. The pressure distribution ports 18 provide a means for the hydraulic load cell element 15 to flex and thereby conform to the contour of the seat cushion and spring when incorporated into a seat.

CLAIMS:

1. A sensor for sensing the weight of an occupant on a vehicle seat, comprising:

a. a first bladder constructed from a plurality of sheets of semi-rigid material sealably connected at the periphery thereof so as to form an enclosed first volume, whereby said first bladder is mountable beneath

a cushion of the seat and supportable by a base of the seat;

b. a fluid contained by said first bladder;

c. a pressure sensor operatively coupled to said first bladder for generating a signal responsive to the pressure of said fluid within said first bladder; and

d. a signal processor for calculating the weight of the occupant from said signal.

2. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, further comprising a one or more bladders each constructed from a plurality of sheets of semi-rigid material sealably connected at the periphery so as to form an associated enclosed volume, wherein said one or more bladders are in fluid communication with said first bladder and are mounted beneath the cushion of the seat and supported by the base of the seat.

3. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, wherein said sheets of semi-rigid material are secured to one another at a plurality of locations within said periphery so as to create a plurality of fluid containing zones in fluid communication with one another.

4. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 3, wherein said plurality of locations at which said sheets of semi-rigid material are secured to one another enable said fluid containing zones to hinge with respect to one another.

5. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 4, wherein said pressure sensor is in fluid communication with the fluid in said first bladder, said pressure sensor is responsive to the difference in pressure between the pressure of the fluid within said bladder and the ambient atmospheric pressure, and said fluid is selected from the group consisting of a liquid, a Bingham plastic and a thixotropic material.

6. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, wherein said fluid is a liquid.

7. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, wherein said fluid is selected from the group consisting of a Bingham plastic and a thixotropic material.

8. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, wherein said pressure sensor is responsive to the difference in pressure between the pressure of the fluid within said first bladder and the ambient atmospheric pressure.

9. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, wherein said pressure sensor is responsive to the strain in the surface of said first bladder.

10. A sensor for sensing the weight of an occupant on a vehicle seat as recited in claim 1, wherein said pressure sensor is in fluid communication with the fluid in said first bladder.

11. A system for sensing the weight of an occupant on a vehicle seat and for controlling a safety restraint system responsive thereto, comprising:

a. a bladder constructed from a plurality of sheets of semi-rigid material sealably connected at the periphery thereof so as to form an enclosed volume, whereby said bladder is mountable beneath a

cushion of the seat and supportable by the base of a seat;

b. a fluid contained by said bladder;

c. a pressure sensor operatively coupled to said bladder for generating a signal responsive to the pressure of said fluid within said bladder; and

d. a signal processor for calculating the weight of the occupant from said signal for generating a control signal for controlling the safety restraint system responsive to said weight measurement.